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STUDY OF THE MIXTURE DISTRIBUTION OF A DOUBLE-ROW RADIAL AIRCRAFT ENGINE

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NACA

WASHINGTON

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ADVANCE RESTRICTED REPORT

STUDY OF THE MIXTURE DISTRIBUTION OF A DOUBLE-ROW RADIAL AIRCRAFT ENGINE

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SUMMARY

A test-stand investigation was conducted to determine the mixture distribution of a conventional double-row radial engine for a wide range of engine speeds, powers, and fuel-air ratios and to evaluate its effect on engine performance. The fuel was injected into the combustion-air stream by a rotating spray ring located near the impeller entrance. Fuel-air ratios of the individual cylinders were determined from chemical analyses of the exhaust gas.

The results exhibited wide variations among the fuel-air ratios of individual cylinders for high over-all fuel-air ratios but comparatively small variations for lean-mixture operation. Engine speed and power exerted only slight influence on the difference between the fuel-air ratios of the richest and the leanest cylinders, although the distribution patterns underwent considerable geometric change with engine speed. For operation in high supercharger gear ratio marked improvement was shown for all fuel-air ratios. Large variations in mixture-distribution patterns resulted from changes in throttle setting, although no definite reduction in spread was observed at any setting. High combustion-air temperatures effected measurable improvement in mixture distribution.

The effects of nonuniform distribution on cooling requirements, fuel consumption, and power have been evaluated assuming uniform distribution of charge air and cooling air. Calculations indicated that in some cases the cylinder-temperature variation resulting from only the measured nonuniform distribution of fuel-air ratio might necessitate an increase of cooling-air pressure drop as much as 28 percent above that required with uniform distribution at the same over-all fuel-air ratio. If the fuel-air ratio of the engine is limited by that of the leanest cylinder, the nonuniform distribution at cruising conditions will necessitate average enrichments as great as 10 percent at rich mixtures with a corresponding increase of 17 percent in brake specific fuel consumption. This enrichment

decreases with a reduction in average fuel-air ratio. For a given over-all fuel-air ratio neither the over-all brake specific fuel consumption nor the total brake horsepower was materially affected by nonuniform distribution.

INTRODUCTION

The trend toward high specific outputs for large internal-combustion engines has accentuated the necessity for avoiding nonuniform mixture distribution and its attendant undesirable effects on fuel economy, cooling requirements, detonation limits, and general engine operation. The increase in fuel consumption and the reduction in mean effective pressure has been found appreciable (reference 1) when mixture distribution is particularly poor. Problems of a more serious nature arise in connection with cylinder cooling and limiting of the maximum power by detonation where, under normal conditions, cylinders receiving mixtures that are considerably leaner than the average may develop unusually high temperatures and approach the point of detonation. The restriction thus imposed (reference 2) may become severe at high powers where proper cooling and detonation suppression may be effected only by the use of slightly overrich mixtures.

The knowledge of mixture distribution in radial engines and of the factors affecting it is incomplete at the present time, owing largely to the difficulties involved in determining the fuel-air ratio of each cylinder. The basic method of measuring the weights of both fuel and air entering each cylinder is, of course, inapplicable to a multicylinder engine with a common fuel supply. The use of temperature variations between cylinders as an index of the mixture distribution (reference 3) is subject to error in the case of an air-cooled engine because of temperature variations caused by factors other than fuel-air ratio. Methods of determining the fuel-air ratio through analysis of the exhaust gas (reference 4) have proved substantially accurate and have been used to determine the mixture distribution in a nine-cylinder radial engine (reference 5) for several conditions. This method, together with the improved technique of exhaust-gas sampling set forth in reference 6, was used in the present investigation.

In the present mixture-distribution study an attempt was made to cover adequate ranges of all operating variables, except fuel volatility (reference 7), that might affect distribution among the cylinders. The effect of mixture distribution on engine performance was evaluated from tests conducted at the Cleveland laboratory of the NACA. The tests were not extended to the range where engine operation was actually restricted by cooling and detonation limitations.

EQUIPMENT AND TEST METHODS

The engine used for the mixture-distribution study was installed on a test stand and fitted with a flight cowling as shown in figure 1. The engine is an 18-cylinder, double-row radial, air-cooled engine with a normal rating of 1500 brake horsepower at 2400 rpm and a take-off rating of 1850 brake horsepower at 2600 rpm. The engine has a single-stage, gear-driven supercharger with low and high gear ratios of 7.6 and 9.45, respectively. One type of fuel, complying with AN-F-28 specifications, was used in all tests. The fuel used had a 90-percent distillation temperature varying from 276° F to 280° F and an average hydrogen-carbon ratio of 0.166.

The test cell was equipped with a bellmouth accommodating the 13-foot, 6-inch, three-bladed propeller. This arrangement reduced the counterflow around the propeller tips as well as the resulting variation in load that appeared to be of considerable magnitude in preliminary tests. Cooling air was supplied through a 42-inch circular duct that discharged 4 feet from the cowling entrance. For tests in which the throttle angle and the combustion-air temperature were varied, the carburetor was connected directly to an external combustion-air system. With this arrangement it was possible to obtain accurate control of the combustion-air temperature and pressure at the carburetor deck.

Carburetor and injection system. - An injection carburetor was used during the mixture-distribution tests. This carburetor (fig. 2) meters the fuel according to the difference between impact and static pressures measured in the main and the boost venturis, respectively. The fuel flows from the carburetor to the slinger ring by way of the fuel-transfer passage and is radially ejected (fig. 3) into the combustion-air stream. The slinger ring turns at impeller speed.

The carburetor incorporates a manual mixture control with full-rich, automatic-rich, automatic-lean, and idle cut-off settings. Inasmuch as intermediate positions of the manual mixture control provide rather coarse adjustment of the mixture strength, an adjustable atmospheric bleed was installed on the suction side of the air diaphragm to permit vernier control. Decreased suction and a consequent reduction of the fuel-air ratio resulted from opening the bleed valve.

Even at the full-rich setting a standard carburetor will not deliver extremely rich mixtures for air flows between 4000 and 9000 pounds per hour, approximately corresponding to brake horsepowers between 600 and 1300. Inasmuch as the tests required a complete range of fuel-air ratios for all powers, the automatic-rich fuel-metering orifice was enlarged to permit richer mixtures over the entire operating range. With this arrangement, it was possible to obtain fine adjustment of the over-all fuel-air ratio for the entire range from 0.050 to 0.115.

Method of determining fuel-air ratios. - During this investigation fuel-air ratios of the individual cylinders were determined from a chemical analysis of the exhaust gas. It has been shown (reference 4) that for a given engine a definite relation exists between the carbon-dioxide content of the exhaust gas and the fuel-air ratio, depending only on the hydrogen-carbon ratio of the fuel. In order to obtain the curve used during the present investigation, an analysis was made of both the normal (unoxidized) and the oxidized exhaust from different cylinders at various engine speeds and powers. From these data, the percentage carbon dioxide in the normal exhaust was plotted against fuel-air ratio (fig. 4) as determined by the carbon-dioxide content of the oxidized exhaust. The relation between the percentage carbon dioxide in the oxidized exhaust and the fuel-air ratio may be computed when the hydrogen-carbon ratio of the fuel is known. The oxidized-exhaust method is fully described in reference 4.

The over-all fuel-air ratio was calculated as the average of the individual fuel-air ratios. Although this method is not precise unless the charge-air weight flow is identical for each cylinder, the errors incurred are well within the experimental accuracy.

Method of sampling. - Samples of exhaust gas were obtained from each cylinder through stainless-steel tubes of 1/4-inch diameter (fig. 5) located in the stack immediately downstream of the exhaust port. The intake end of each tube was flattened to form a slot 0.01 inch wide of the type recommended in reference 6 for reducing contamination. Copper tubing, 1/4 inch in diameter and 30 feet in length, led from the sampling tubes to a set of water traps provided to prevent blocking of the lines with condensate. From the water traps the lines were extended to the sampling bottles by plastic tubing 15 feet in length. A schematic diagram of the sampling arrangement is shown in figure 6.

Exhaust-gas samples were collected in 300-milliliter bottles provided with a tight-sealing stopcock at each end. It was found that the pressure built up in the stack by use of the collector ring, in addition to the impact pressure of the exhaust gas, was sufficient to force the gas through the sampling lines and the glass pipettes at a reasonably high rate. Preliminary tests showed that, by purging the bottles with exhaust gas for a period of 5 minutes, it was possible to obtain an undiluted sample. This self-scavenging method of collecting samples provided undiluted samples under all test conditions and required a minimum of attention and manipulation.

Investigation of exhaust dilution. - Because the engine was equipped with a collector ring, tests were conducted (reference 8) to determine whether the samples obtained from individual stacks were characteristic of that particular cylinder or whether, because of infiltration of exhaust gas from the collector ring, they represented the average for a number of cylinders. The tests were based on the fact that an undiluted exhaust-gas sample taken from a cylinder that is not firing contains no carbon dioxide but only an unburned mixture of air and gasoline vapor. With the assumption that all the carbon dioxide found in the exhaust could be attributed to infiltration of burned exhaust from adjacent cylinders, the magnitude of dilution for a given cylinder could be measured from analysis of an exhaust sample obtained when the ignition current for that cylinder was short-circuited. These tests, conducted at various engine speeds and powers, showed the maximum dilution to be less than 5 percent. Calculations indicate that a dilution of 5 percent produces an error of only 0.001 in the fuel-air ratio, which is within the limit of accuracy of measurements. The use of a collector ring in all subsequent mixture-distribution tests was therefore considered justifiable.

TESTS AND RESULTS

For a fuel of a given volatility, the principal independent operating variables affecting the mixture distribution in a given engine are the over-all fuel-air ratio of the engine, the engine speed, the supercharger gear ratio, the charge weight flow or the engine power, the throttle setting, and the combustion-air inlet temperature. The test conditions (table I) were arranged to investigate each of these variables in a manner simulating actual operation. For tests in which the over-all fuel-air ratio, the engine speed, the power, or the intake-air temperature was varied, the carburetor-deck pressure was maintained atmospheric and the throttle setting was varied in order to maintain fixed power. These results therefore correspond to normal sea-level operation and show the effect of both the particular variable under investigation and the associated variation of throttle setting. For tests in which the engine power and the engine speed were varied, the variation in throttle angle necessitated by atmospheric carburetor-deck pressure is one of the most important influences on mixture-distribution changes. Tests of varying throttle setting during which the carburetor-deck pressure was changed to maintain constant power simulate operation at altitude.

Variation of mixture distribution with over-all fuel-air ratio. - Changes in over-all fuel-air ratio result in large variations of mixture distribution, as shown in figures 7 and 8. The distribution improves appreciably as the over-all mixture becomes leaner than

0.100. For example, the difference between the richest and the leanest cylinders (fig. 7(a)) is decreased from 0.032 at a fuel-air ratio of 0.101 to 0.003 at a fuel-air ratio of 0.059. This trend, although not of the same magnitude, prevails at all operating conditions and apparently results from the more nearly complete fuel evaporation at low fuel flows where the irregularities of distribution caused by the concentration of fuel droplets are reduced.

Variation of mixture distribution with engine speed. - For operation with low supercharger gear ratio, the variation of mixture strength between the richest and the leanest cylinders is not appreciably affected by changes of engine speed between 1600 and 2400 rpm (fig. 9) despite the accompanying variation in throttle angles. The shape of the distribution patterns, however, varies markedly. At rich mixtures (figs. 9(b) and 9(d)), where the most uneven distribution occurs, two well-defined peaks may be observed occurring in the neighborhood of cylinders 10 and 16. As the engine speed increases, the peak near cylinder 16 is diminished whereas that near cylinder 10 becomes more predominant. At high speeds each of the maximum points moves to the adjacent cylinder in a direction opposite that of impeller rotation; that is, from cylinders 10 and 16 to cylinders 8 and 14, respectively.

Variation of mixture distribution with supercharger gear ratio. - When the engine was operated in high supercharger gear ratio, the mixture distribution (fig. 10) was greatly improved as compared with the distribution occurring at similar conditions for operation with low supercharger gear ratio. No improvement of mixture distribution was observed, however, when the impeller speed increased as a result of increased engine speed. The more nearly uniform mixture distribution at high impeller speeds may result from the higher combustion-air temperature and, consequently, the better evaporation of the fuel passing through the supercharger as well as from the more thorough mixing at the diffuser entrance. The lack of improvement in mixture distribution when the engine and the impeller speeds vary in direct ratio suggests that the effect of increased engine speed may be to nullify the improvement due to increased impeller speed.

Variation of mixture distribution with power. - Results of tests at various values of brake horsepower are presented in figure 11 for three over-all fuel-air ratios. The range of mixture strengths among individual cylinders for a given over-all fuel-air ratio is not appreciably affected by changes in power, as can be seen from the similar distribution patterns obtained for the range of powers covered. The variation of throttle angle required to produce the power changes appears to have little influence on the mixture distribution.

Variation of mixture distribution with throttle setting. - Mixture-distribution tests were conducted at throttle settings required by atmospheric carburetor-deck pressure, wide-open throttle, and at one intermediate setting for several engine powers and speeds and for two values of over-all fuel-air ratio. The results of the tests with normal and wide-open throttle are shown in figure 12. The mixture-distribution patterns obtained with the intermediate throttle setting were omitted from the graphs because they were found to approximate the average of those obtained with the other two settings. Large variations in the patterns with change in throttle setting were especially noticeable at the low powers where the angle between normal and wide-open throttle setting is the greatest. At a brake horsepower of 800, an engine speed of 2200 rpm, and an over-all fuel-air ratio of 0.10, cylinder 14 changed from the leanest cylinder (fuel-air ratio of 0.09) at normal throttle angle to the richest one (fuel-air ratio of 0.115) at wide-open throttle. Although the patterns changed considerably with throttle angle no definite trend of improvement in mixture distribution was observed. The cylinder temperatures associated with the mixture distribution shown in figure 12(a) are presented in figure 13 and indicate the changes in temperature pattern that largely result from the changes in mixture distribution.

Variation of mixture distribution with combustion-air temperature. - Reference 7 has shown that variations of mixture distribution occur for large changes in combustion-air temperature. For the present investigation, the carburetor-deck temperature was varied from about 40° F to 135° F. The results of figure 14 indicate that, although the shape of the mixture-distribution patterns was not affected, the spread in fuel-air ratio was appreciably reduced as the combustion-air temperature was increased. At a temperature of 134° F the spread was less than 0.016. The improvement in distribution at the higher combustion-air temperatures can be attributed to increased fuel evaporation resulting in better mixture of the fuel and air.

Reproducibility and accuracy. - Distribution patterns taken at comparable engine conditions for the different periods of the test program show good agreement (fig. 15) and, in general, the reproducibility of results was most satisfactory. Inasmuch as all of the tests except those of varying combustion-air temperature and of varying throttle setting were conducted with an atmospheric induction system, the combustion-air temperature was fixed by ambient atmospheric conditions. The variations were, however, of such small magnitude that the errors incurred were of little importance.

DISCUSSION

In the following discussion, uniform distribution of charge air and cooling air is assumed. Although this assumption is not strictly true for any engine, it serves to isolate the effect of nonuniform mixture distribution and to demonstrate its influence on engine performance.

The symmetry of the impeller-diffuser combination apparently affords little opportunity for generation of a systematically irregular mixture distribution within the supercharger except by action of gravitational forces. The characteristics of the mixture-distribution patterns, therefore, must either be formed upstream of the impeller or result from liquid fuel draining to the bottom of the supercharger front shroud or the supercharger collector. The marked response of the mixture distribution to such a factor as throttle setting indicates that the velocity profile of the combustion air entering the impeller is a probable source of poor mixture distribution. It is possible that nonuniform velocity profile caused by flow separation from the elbow walls, the turning vanes, and the throttle will introduce concentration of fuel droplets near the points at which the low-velocity air enters the impeller. If the velocity profile at the impeller inlet could therefore be made more nearly uniform, corresponding improvement in mixture distribution should result. The foregoing results, although obtained below the maximum power range, may be used to determine the restrictions imposed by the nonuniform distribution and to ascertain what degree of improvement might be expected by equalizing the fuel-air ratio for each cylinder.

Effect of mixture distribution on fuel consumption. - Operation with uneven distribution of fuel among the cylinders of a multicylinder engine results in inherently less efficient utilization of fuel than does operation with uniform distribution if the other engine conditions remain fixed. Owing to the nature of the relation between brake specific fuel consumption and fuel-air ratio, the over-all brake specific fuel consumption for nonuniform mixture distribution is necessarily greater than that for uniform distribution; the magnitude depends on the degree of nonuniformity of mixture distribution and the over-all fuel-air ratio.

If the fuel-air ratio for an individual cylinder is $(F/A)_i$, the corresponding brake specific fuel consumption is $bsfc_i$ (fig. 16), the charge-air weight flow $\frac{W_a}{n}$ is equal for each cylinder, then the brake horsepower of any cylinder bhp_i may be expressed

$$bhp_i = \frac{\frac{W_a}{n}(F/A)_i}{bsfc_i}$$

where n is the number of cylinders. The power of the entire engine is then given by the summation of powers for the individual cylinders.

$$bhp_{total} = \frac{W_a}{n}\sum_{i=1}^{n}\frac{(F/A)_i}{bsfc_i}$$

By the same reasoning the total fuel flow W_f in pounds per hour may be expressed as

$$W_f = \frac{W_a}{n}\sum_{i=1}^{n}(F/A)_i$$

Thus, by definition, the over-all brake specific fuel consumption $bsfc_o$ becomes

$$bsfc_o = \frac{\sum_{i=1}^{n}(F/A)_i}{\sum_{i=1}^{n}\frac{(F/A)_i}{bsfc_i}}$$

If the mixture distribution were uniform, the fuel-air ratio for each cylinder would equal that of the engine average and the brake specific fuel consumption of the entire engine would be the same as that of each individual cylinder.

Calculations for the poorest observed distribution show negligible increases in over-all brake specific fuel consumption above that corresponding to uniform distribution for the same over-all fuel-air ratio. This small variation may be explained by the fact that at high fuel-air ratios the over-all brake specific fuel consumption is not sensitive to moderate changes of mixture strength among cylinders and at leaner mixtures the actual distribution of the engine is relatively uniform. On the other hand, if poor distribution had been encountered at lean mixtures, an appreciable effect on the over-all brake specific fuel consumption could have resulted.

When, because of detonation or cooling limitations, the extent to which the over-all fuel-air ratio may be reduced is governed by the leanest cylinder, the effect of nonuniform mixture distribution attains considerable importance. An experimental curve showing the relation between the over-all fuel-air ratio and the fuel-air ratio of the leanest cylinder is presented in figure 17. The dashed line in the figure represents the same relation for uniform mixture distribution. The difference between the ordinates of the two curves for a given value of the abscissa then represents the increase in over-all fuel-air ratio required by nonuniform mixture distribution to attain a given value for the fuel-air ratio of the leanest cylinder.

The data points obtained at various powers and speeds in low-blower operation and at sea-level carburetor-deck pressure and temperature indicate that the enrichment required depends largely on fuel-air ratio. Owing to the improvement of distribution at low mixture strengths, the difference between the over-all fuel-air ratio and that of the leanest cylinder decreases with decreasing mixture strength. The enrichment is decreased from 10 to 8 percent when the over-all fuel-air ratio is reduced from 0.10 to 0.07. It follows from the curves of brake specific fuel consumption at low supercharger gear ratio (fig. 16) that enrichments of 8 and 10 percent correspond to increases in brake specific fuel consumption of 5.6 and 17.0 percent, respectively, at 2000 rpm.

Effect of mixture distribution on engine power. - It has been shown that, disregarding knock limitations, the power available for a given total charge-air weight flow W_a may be expressed

$$bhp_{total} = \frac{W_a}{n} \sum_{i=1}^{n} \frac{(F/A)_i}{bsfc_i}$$

If the over-all brake specific fuel consumption is greater for nonuniform than for uniform distribution, the total brake horsepower is smaller, for a given over-all fuel-air ratio, for nonuniform than for uniform distribution. Inasmuch as nonuniform mixture distribution had only slight effects on the over-all brake specific fuel consumption, it follows from the relation of over-all brake specific fuel consumption to over-all power that effects of the same magnitude will be observed for over-all power. Calculations using mixture-distribution patterns obtained in the present investigation substantiated this conclusion.

Effect of mixture distribution on cooling requirements. - From the cylinder-cooling relations of the type of engine used in these tests (reference 9), the ratio of cooling-air pressure drops required for uniform and nonuniform mixture distribution to cool the cylinder-head temperature to a given value may be expressed

$$\frac{\Delta p'}{\Delta p} = \left[\frac{T_g' - T_h}{T_g - T_h}\right]^{3.1} \qquad (1)$$

where

$\Delta p'$ cooling-air pressure drop required with nonuniform mixture distribution

Δp cooling-air pressure drop required with uniform mixture distribution

T_g' maximum mean effective combustion-gas temperature for engine with nonuniform mixture distribution

T_g mean effective combustion-gas temperature for engine with uniform distribution

T_h rear spark-plug-gasket temperature

An experimental curve of mean effective combustion-gas temperature against fuel-air ratio, obtained from reference 9, is shown in figure 18.

The percentage increase in pressure drop required for proper cooling, as determined by the fuel-air ratio of the leanest cylinder, is shown in figure 19 for various over-all fuel-air ratios and an average cylinder-head temperature of 350° F. Figure 19 is valid for all nonuniform mixture distributions provided that the fuel-air ratio of the leanest cylinder is on the rich side of the theoretically correct mixture.

The dashed line in figure 19 is a cross plot of mixture-distribution data obtained at 800 and 1000 brake horsepower at 2200 rpm. The points on this line represent fuel-air ratios of the leanest cylinder for any given average fuel-air ratio and the corresponding percentage increase in pressure drop as calculated by equation (1). Under these conditions nonuniform distribution causes a 28-percent increase in cooling-air pressure-drop requirement at a fuel-air ratio of 0.085.

SUMMARY OF RESULTS

The results of mixture-distribution tests conducted on a double-row radial aircraft engine, together with the analysis of these results, may be summarized as follows:

1. The variation of mixture strength between the richest and the leanest cylinders showed a marked decrease with reduction of the over-all fuel-air ratio below 0.10; for 800 brake horsepower and 1600 rpm the variation decreased from 0.032 to 0.003 at average fuel-air ratios of 0.101 and 0.059, respectively.

2. A variation of engine speed from 1600 to 2400 rpm had no appreciable effect upon the difference between the fuel-air ratios of the richest and the leanest cylinders although the geometry of the mixture-distribution pattern was greatly affected.

3. Operation at high supercharger gear ratio resulted in a marked improvement of mixture distribution at all fuel-air ratios, powers, and engine speeds.

4. When the engine power was varied at constant speed from 800 to 1500 brake horsepower, neither the variation of mixture strength between the richest and the leanest cylinders nor the shape of the mixture-distribution pattern was appreciably affected.

5. Changes in throttle setting produced large variations in mixture-distribution patterns; however, no definite improvement in distribution was observed at any of the throttle settings tested. This trend is similar to that observed in the variable engine-power and engine-speed tests during which throttle-angle variation was appreciable.

6. Increasing the combustion-air temperature from 45° F to 134° F resulted in an appreciable improvement in mixture distribution.

7. Calculations showed that, if engine operation is limited by the fuel-air ratio of the leanest cylinder, the nonuniform distribution will necessitate enriching the over-all mixture 8 and 10 percent at fuel-air ratios of 0.07 and 0.10, respectively. These enrichments correspond to increases of 5.6 and 17.0 percent, respectively, in the brake specific fuel consumption at 2000 rpm.

8. Evaluation of the test data indicated that with constant charge-air weight flow, neither the brake specific fuel consumption nor the brake horsepower was appreciabley affected by the existing

nonuniform fuel distribution for operation at a given over-all fuel-air ratio and below knock-limited power.

9. Calculations based upon test data showed that the nonuniform fuel distribution may increase the pressure-drop requirement as much as 28 percent over that for uniform distribution, provided that the cooling air and the charge air are evenly distributed.

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Cleveland, Ohio.

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TABLE I

SUMMARY OF CONDITIONS AT WHICH MIXTURE DISTRIBUTIONS WERE OBTAINED

Engine power (bhp)	Engine speed (rpm)	Over-all fuel-air ratio	Supercharger gear ratio	Variable	Figure
800	1600	Varied	Low blower	Fuel-air ratio	7(a)
800	1800	---do---	---do-------	---do-----------	7(b)
800	2000	---do---	---do-------	---do-----------	7(c)
800	2200	---do---	---do-------	---do-----------	7(d)
1000	1800	---do---	---do-------	---do-----------	8(a)
1000	2000	---do---	---do-------	---do-----------	8(b)
1000	2200	---do---	---do-------	---do-----------	8(c)
1000	2400	---do---	---do-------	---do-----------	8(d)
800	Varied	0.070	---do-------	Engine speed	9(a)
800	--do--	.106	---do-------	---do-----------	9(b)
1000	--do--	.074	---do-------	---do-----------	9(c)
1000	--do--	.110	---do-------	---do-----------	9(d)
800	2000	Varied	Varied	Blower ratio	10(a)
1000	2000	---do---	---do-------	---do-----------	10(b)
Varied	2200	.068	Low blower	Brake horsepower	11(a)
Do----	2200	.089	---do-------	---do-----------	11(b)
Do----	2400	.113	---do-------	---do-----------	11(c)
Do----	Varied	.100	---do-------	Throttle setting	12(a)
Do----	--do--	.078	---do-------	---do-----------	12(b)
1000	2200	.100	---do-------	Carburetor-deck temperature	14

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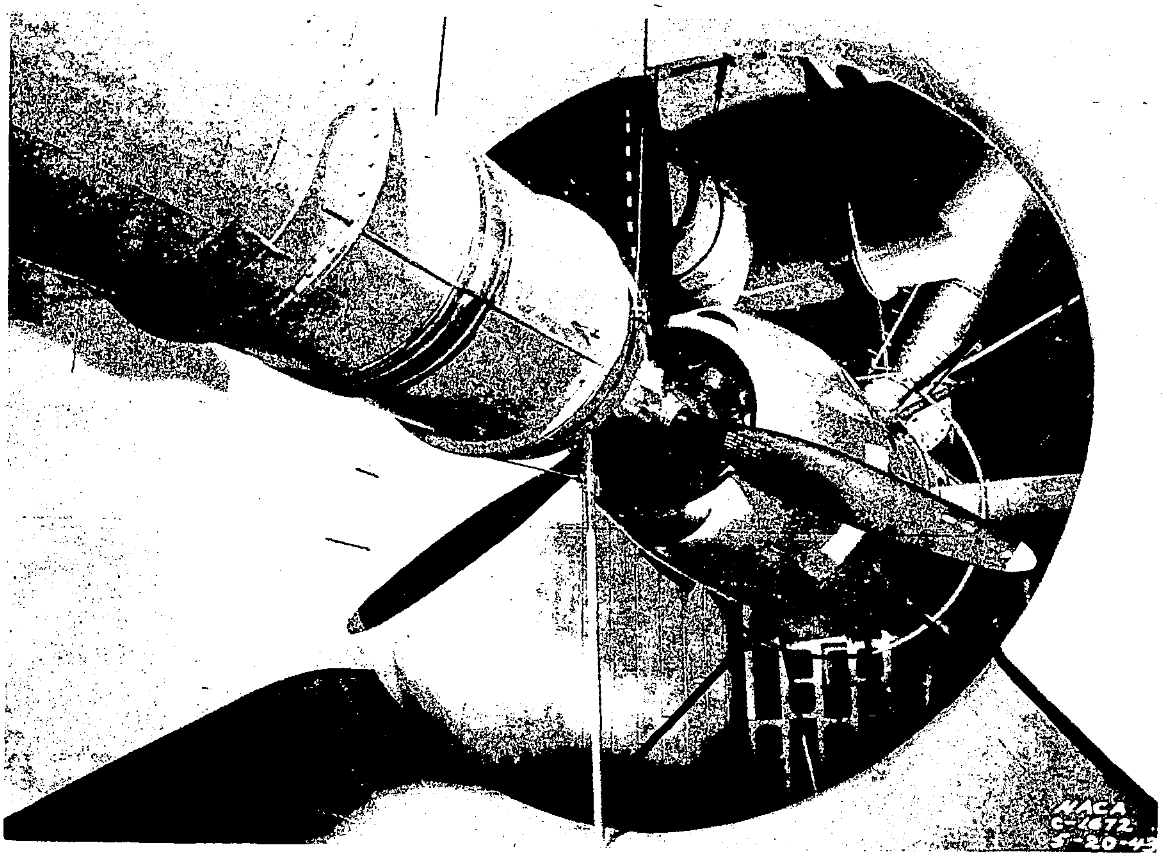

Figure 1. - Installation of test engine and cowling.

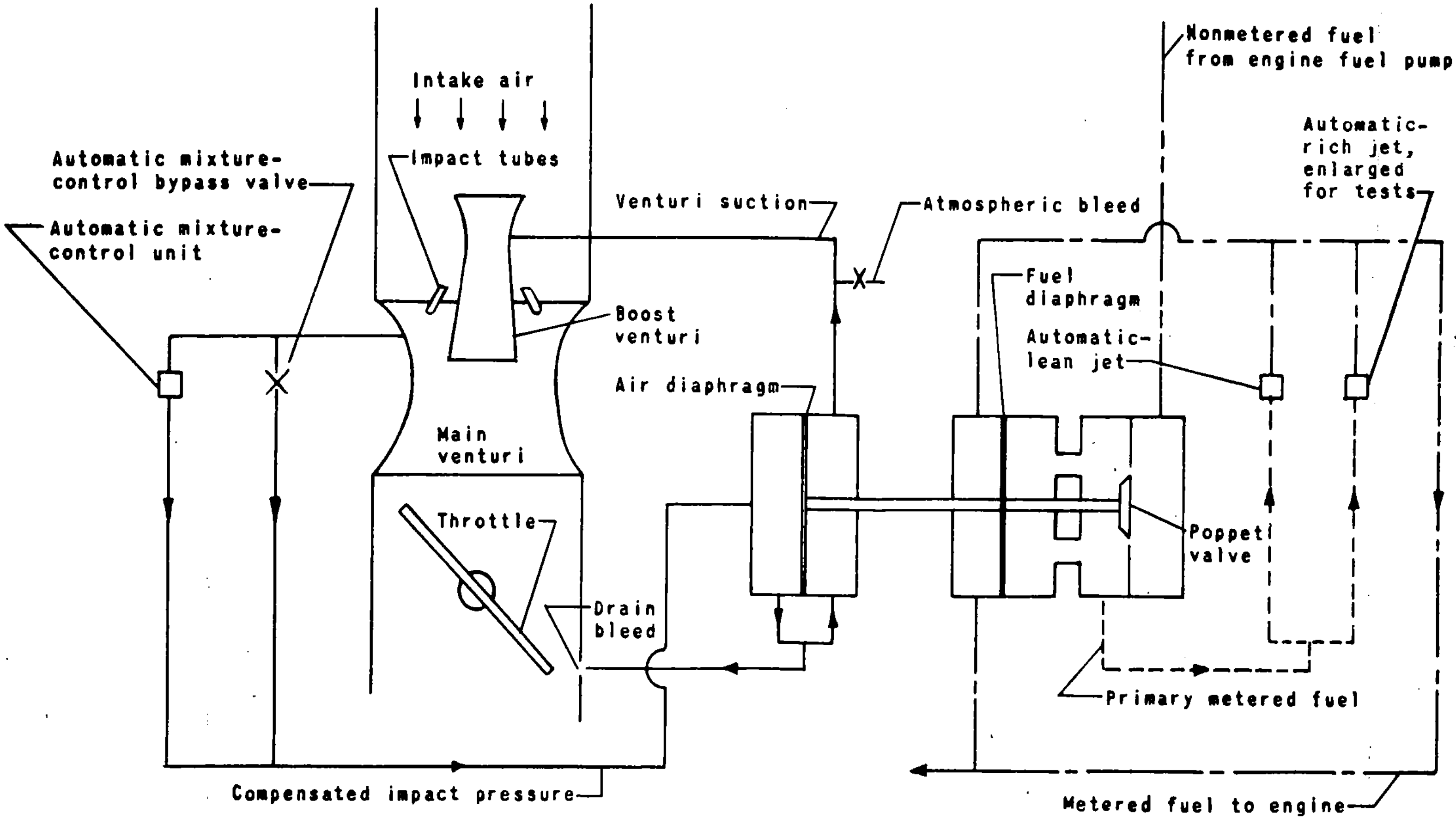

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Figure 2. - Schematic diagram of modified injection carburetor.

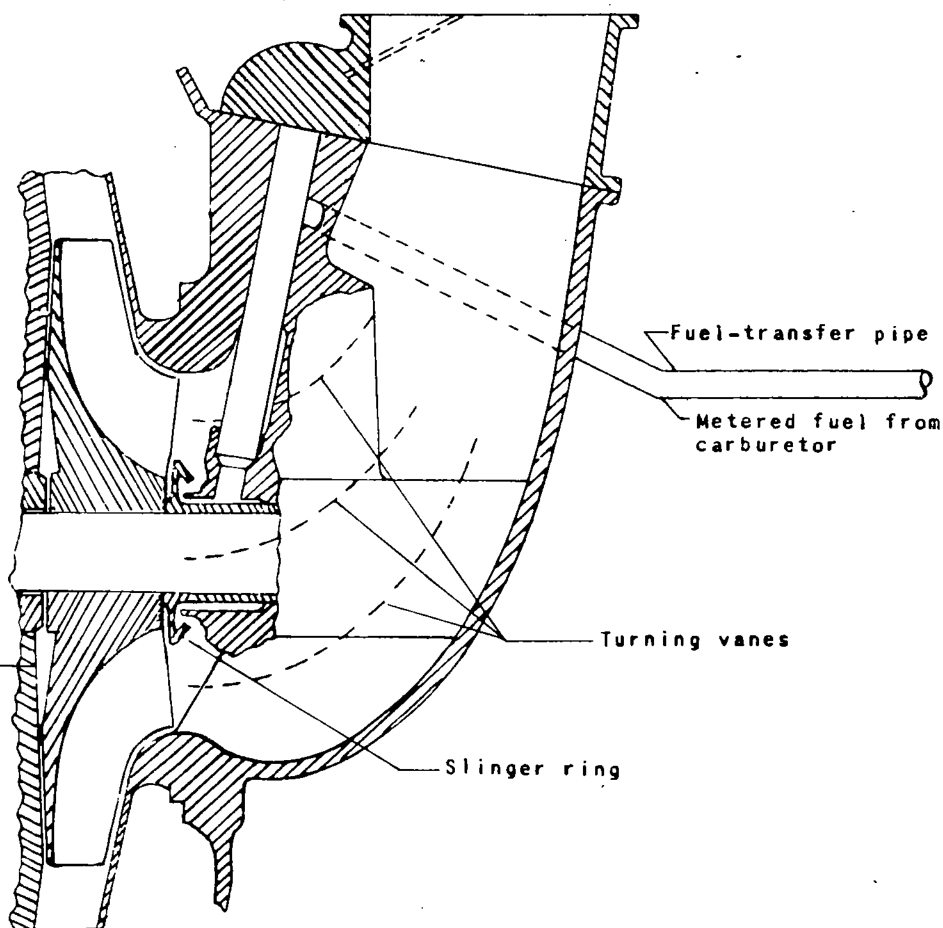


Figure 3. - Schematic diagram of slinger-ring fuel-injection system.

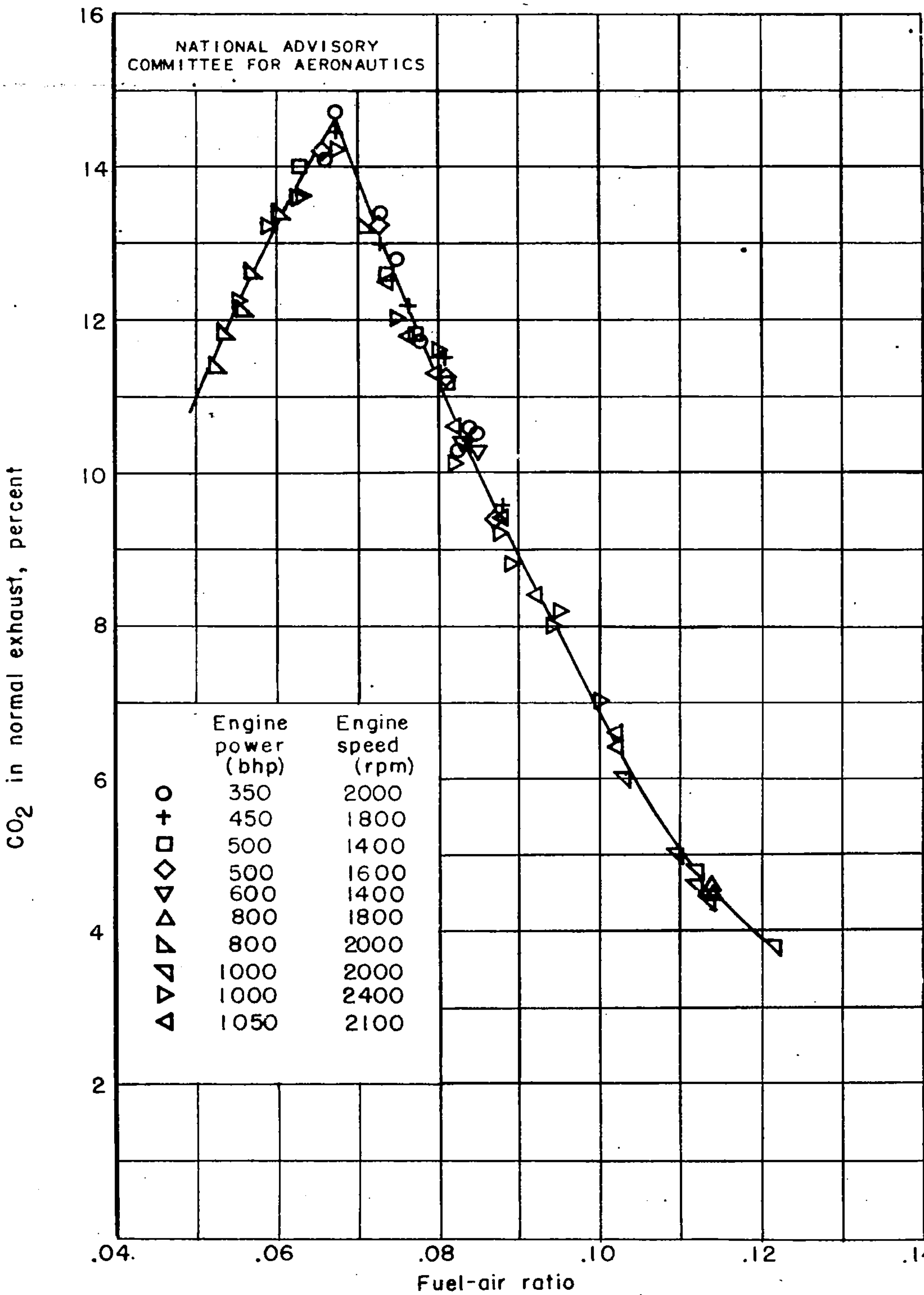


Figure 4.— Relation between fuel-air ratio and carbon dioxide in normal exhaust. Fuel, AN-F-28, performance grade, 130; hydrogen-carbon ratio, 0.166.

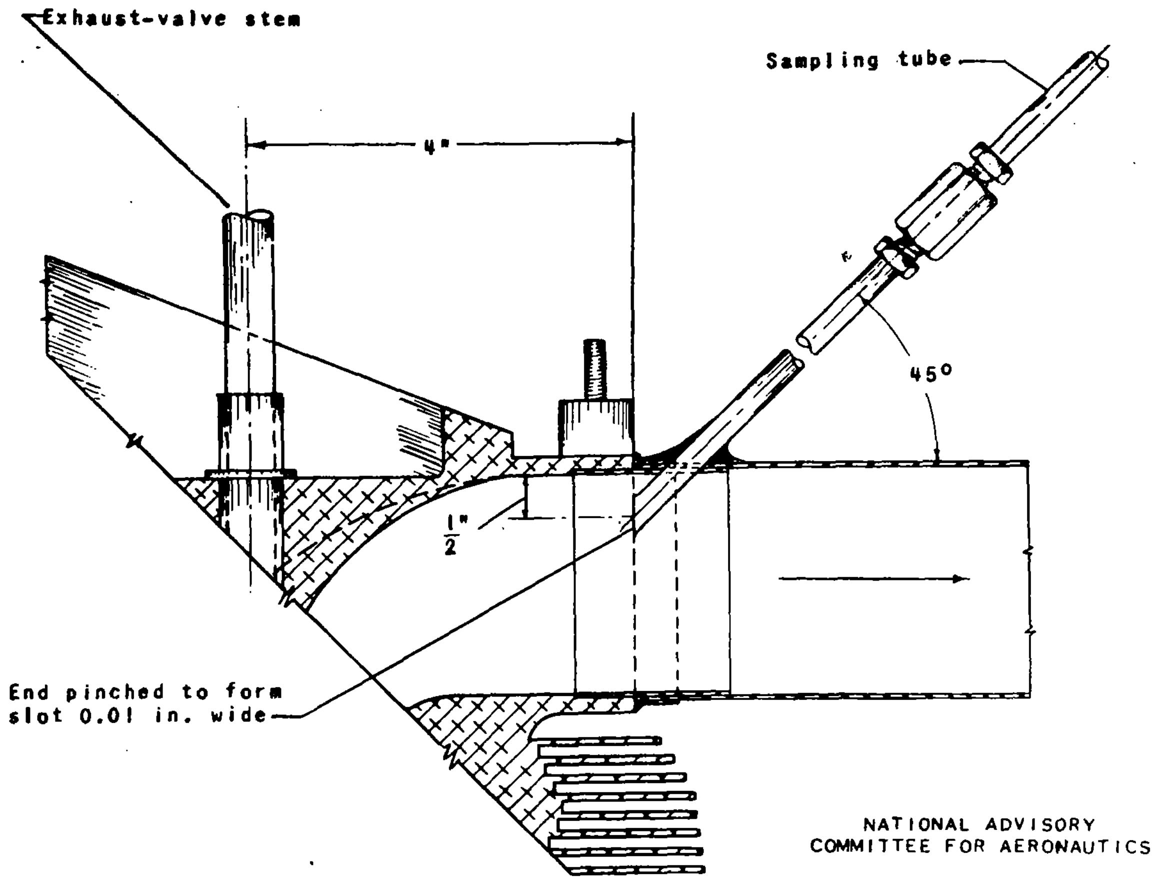


Figure 5. - Location of exhaust-gas sampling tube on each cylinder.

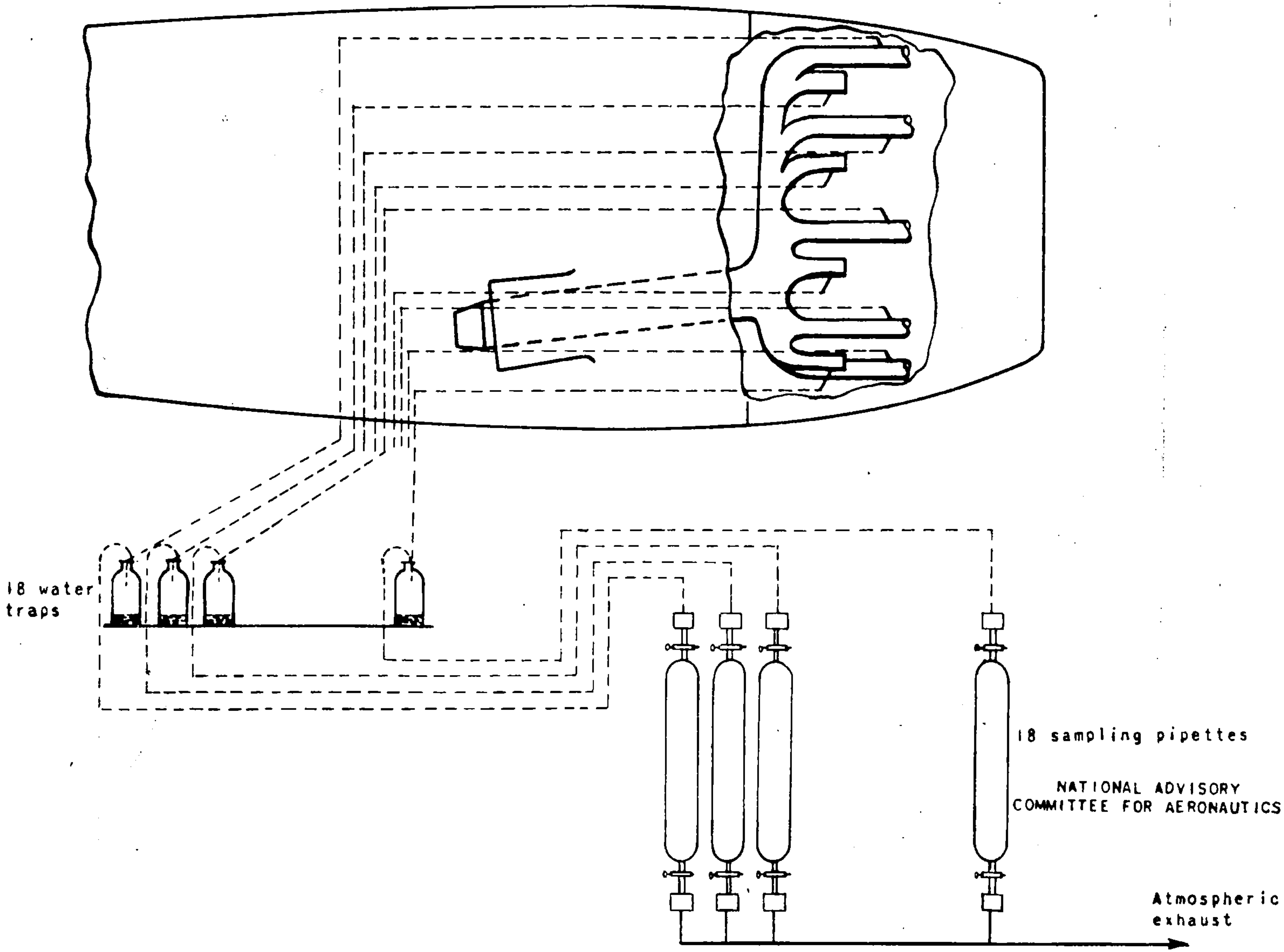
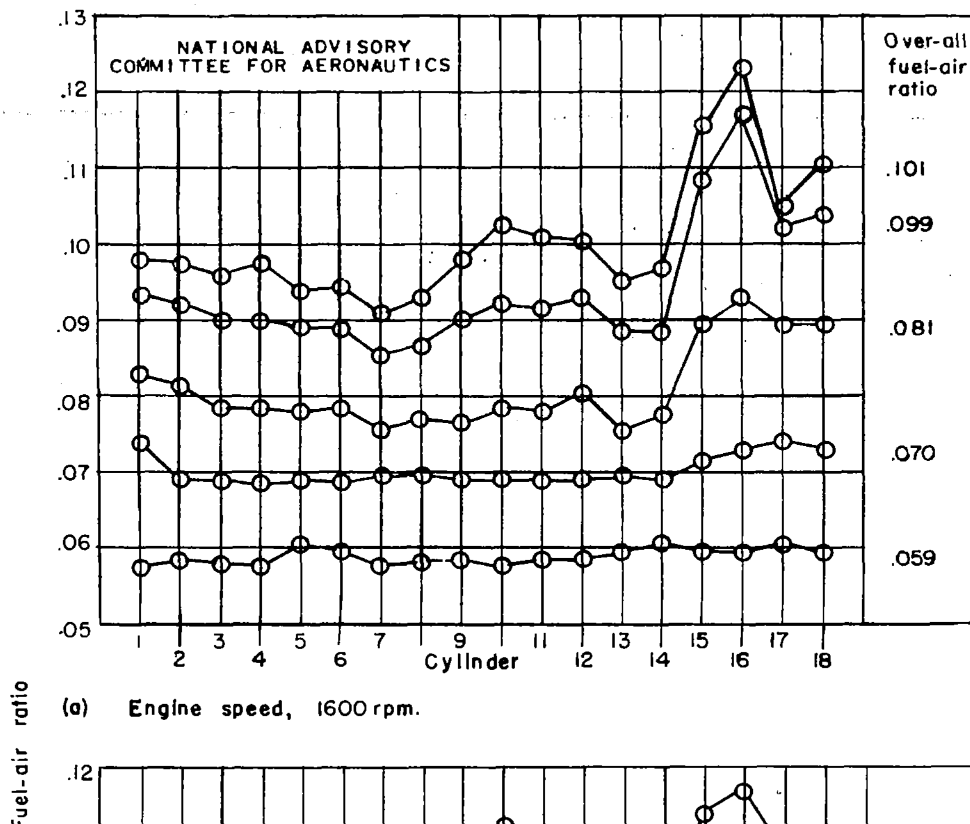
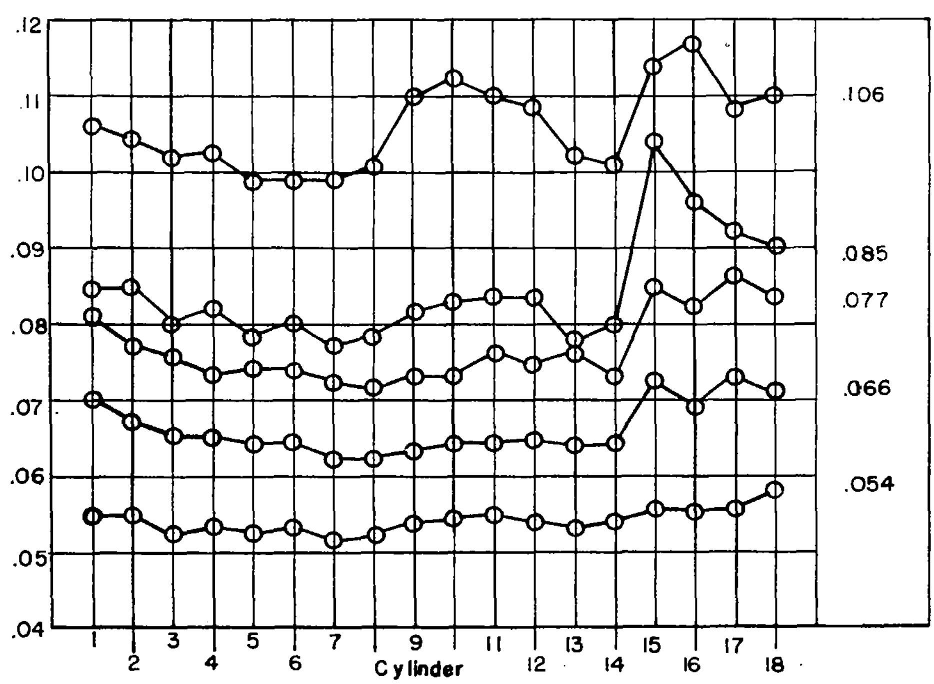


Figure 6. - Schematic diagram of exhaust-gas-sampling system.



(a) Engine speed, 1600 rpm.



(b) Engine speed, 1800 rpm.

Figure 7.— Effect of over-all fuel-air ratio on mixture distribution for 800 brake horsepower and various values of engine speed. Low supercharger gear ratio.

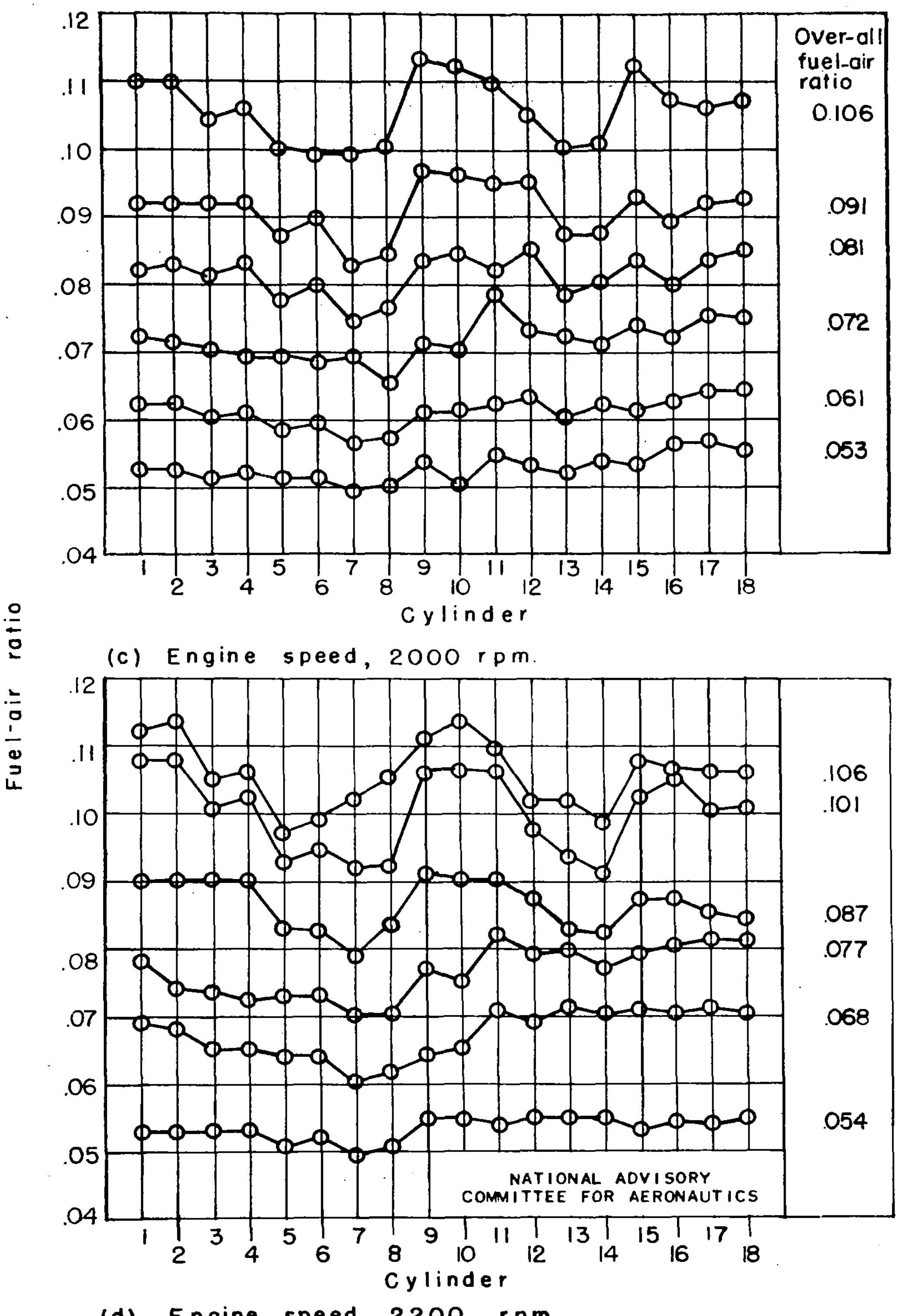


Figure 7. - Concluded.

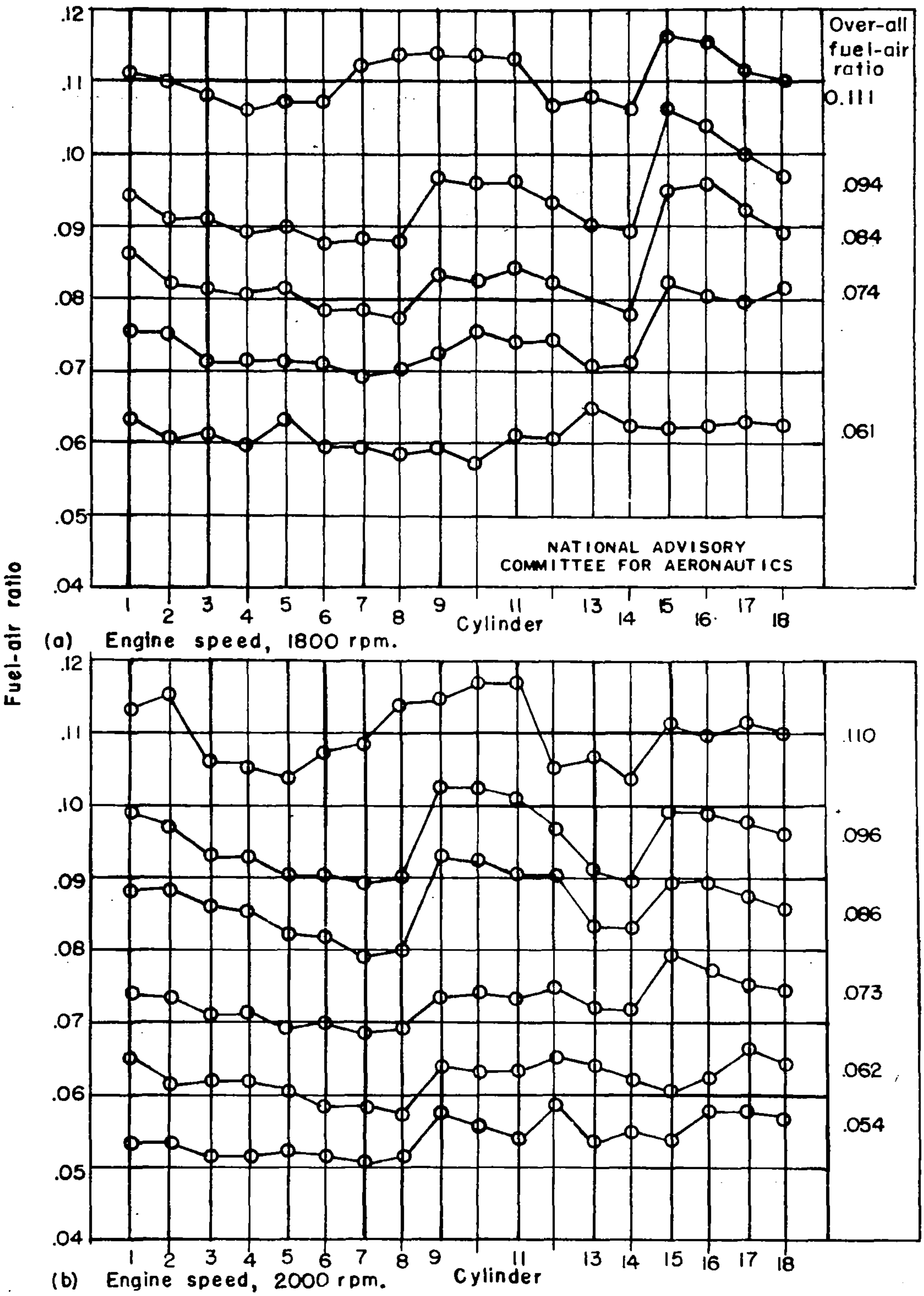
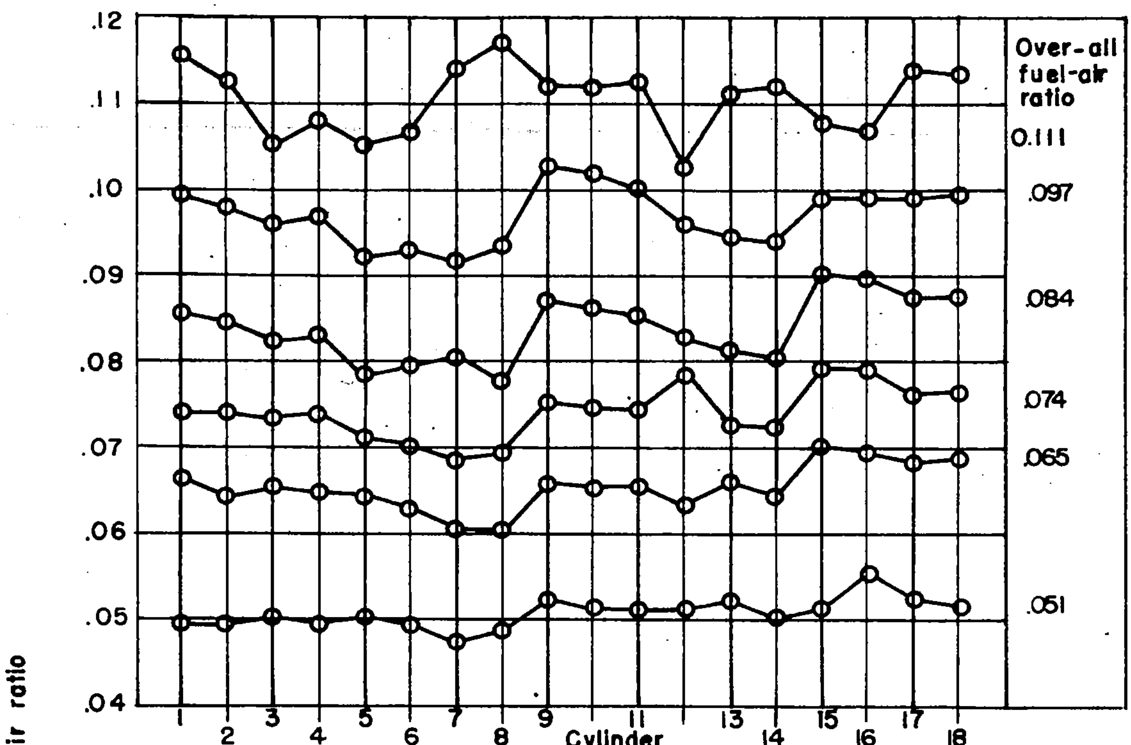
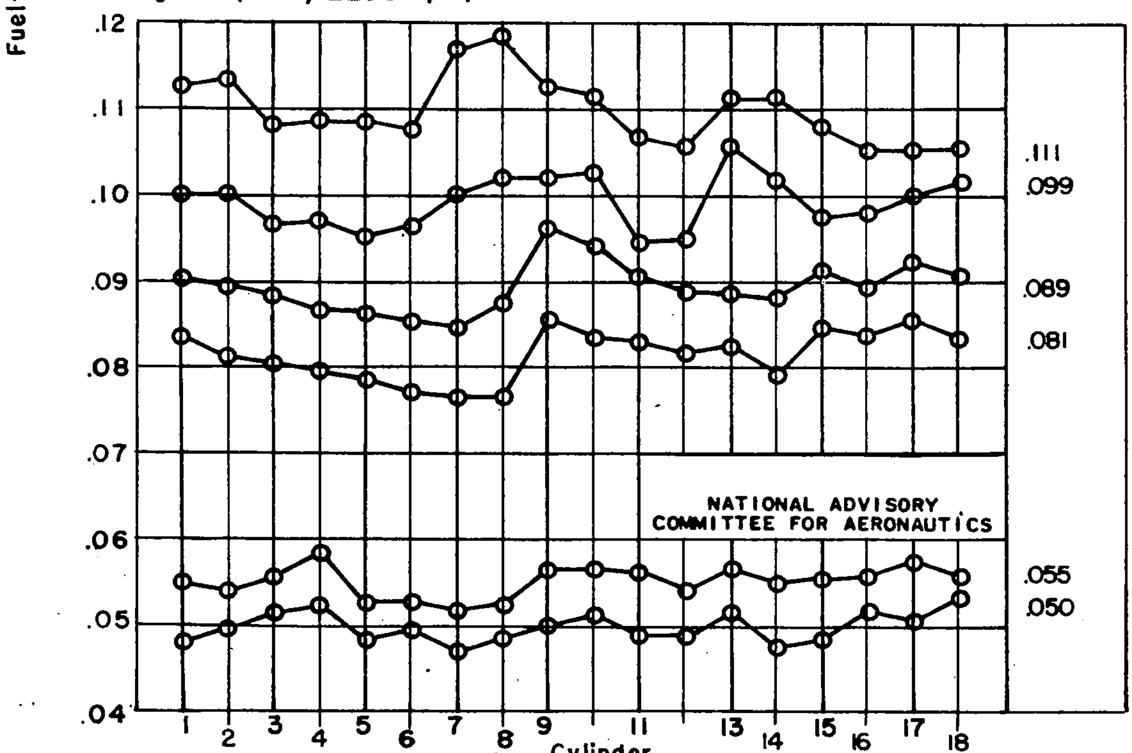


Figure 8.– Effect of over-all fuel-air ratio on mixture distribution for 1000 brake horsepower and various engine speeds. Low supercharger gear ratio.

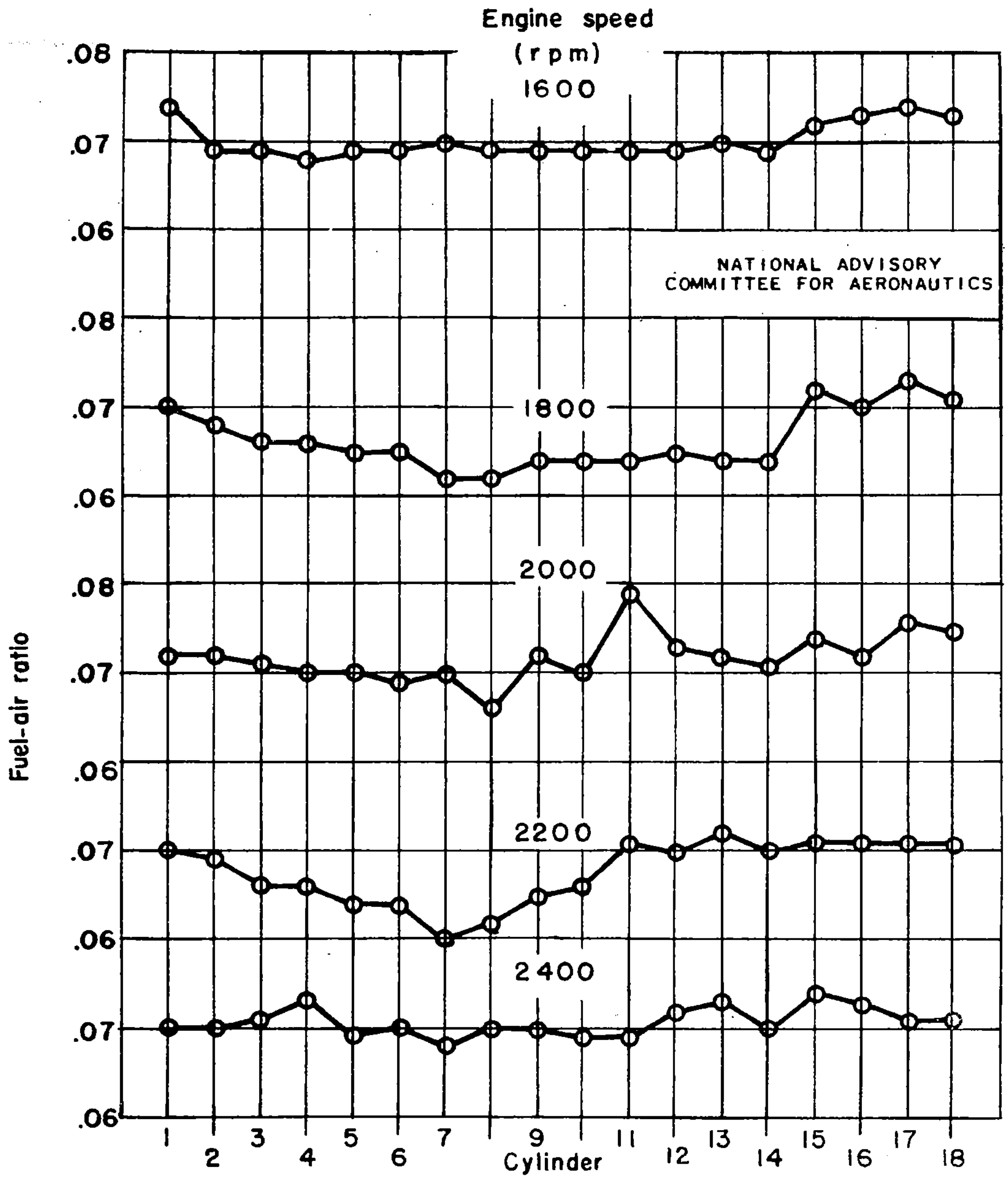


(c) Engine speed, 2200 rpm.



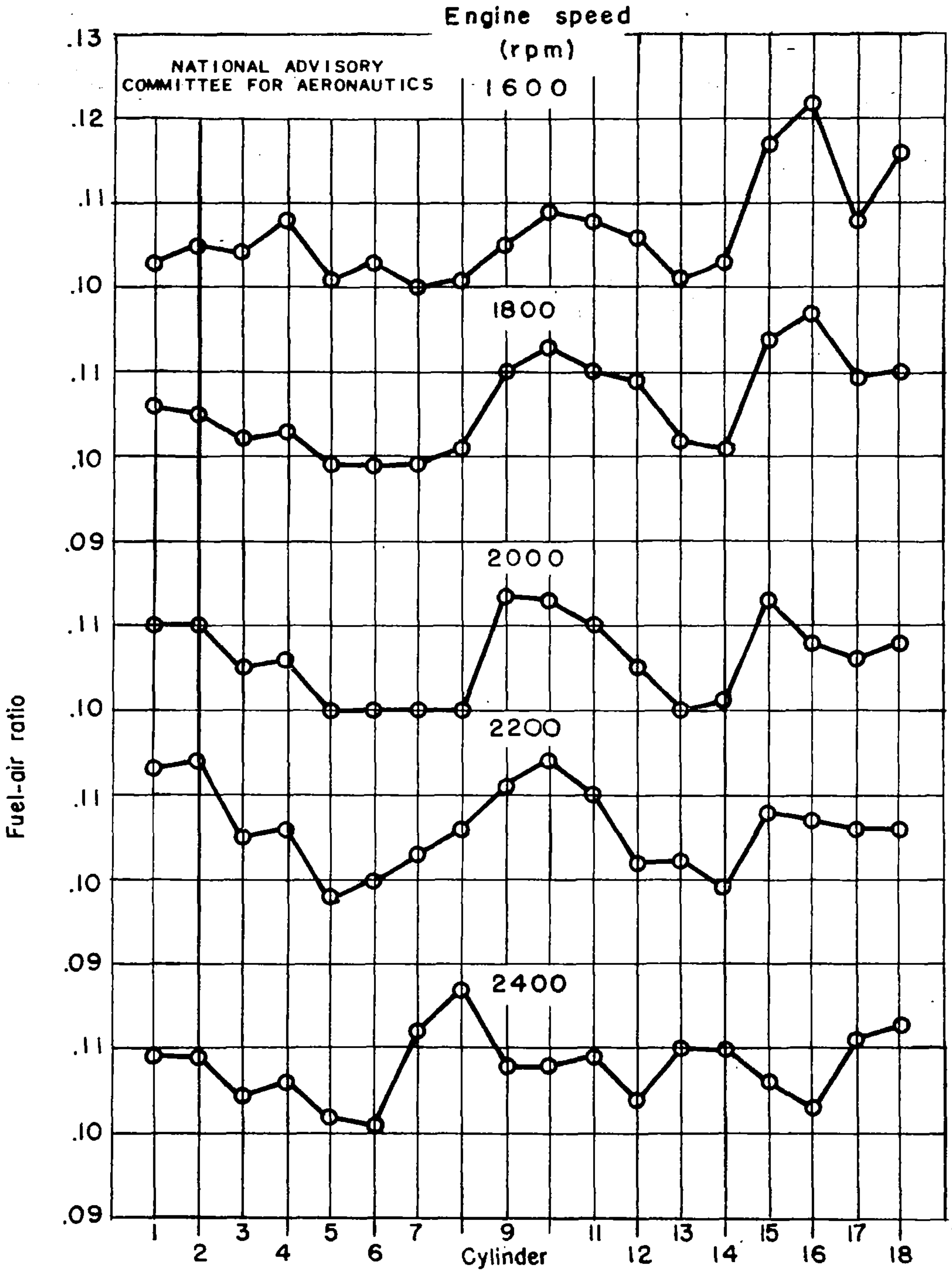
(d) Engine speed, 2400 rpm.

Figure 8 — Concluded.



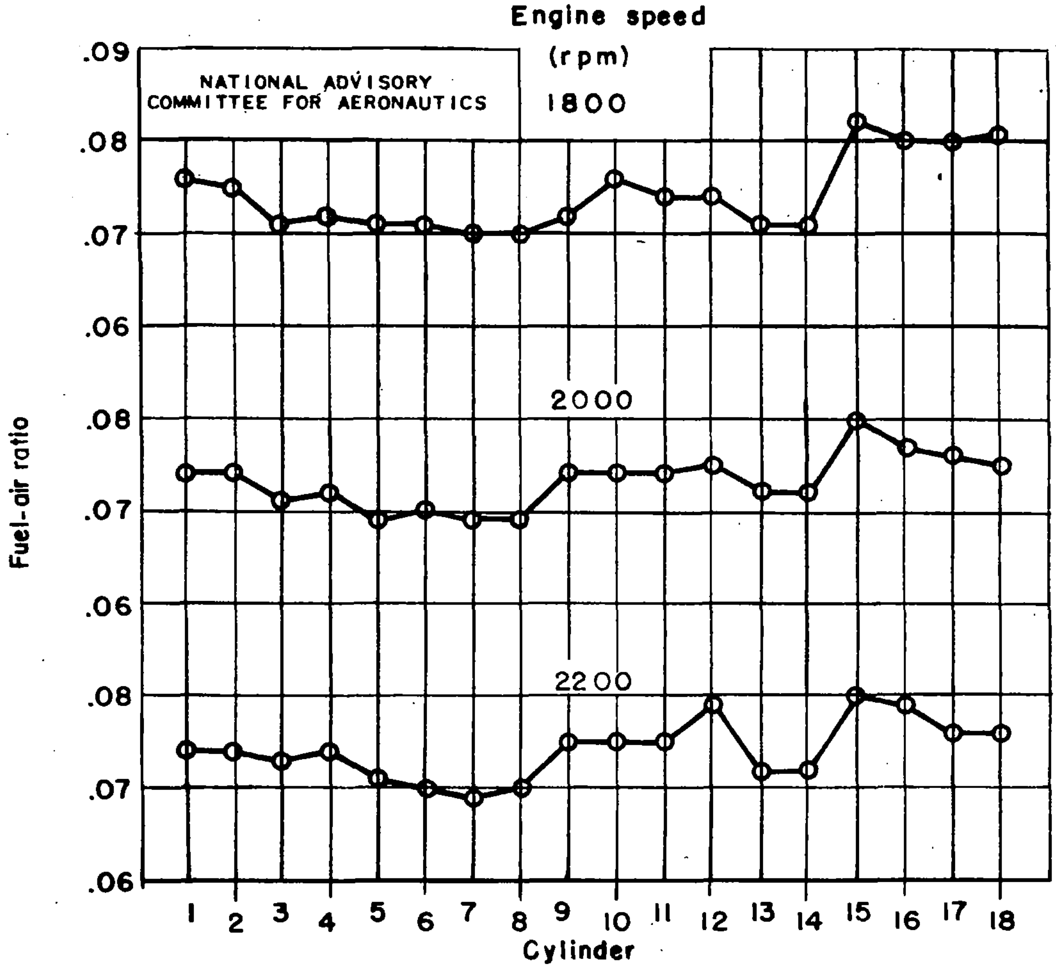
(a) 800 brake horsepower; approximate over-all fuel-air ratio, 0.070.

Figure 9.— Effect of engine speed on mixture distribution for various values of engine power, and over-all fuel-air ratio Low supercharger gear ratio.



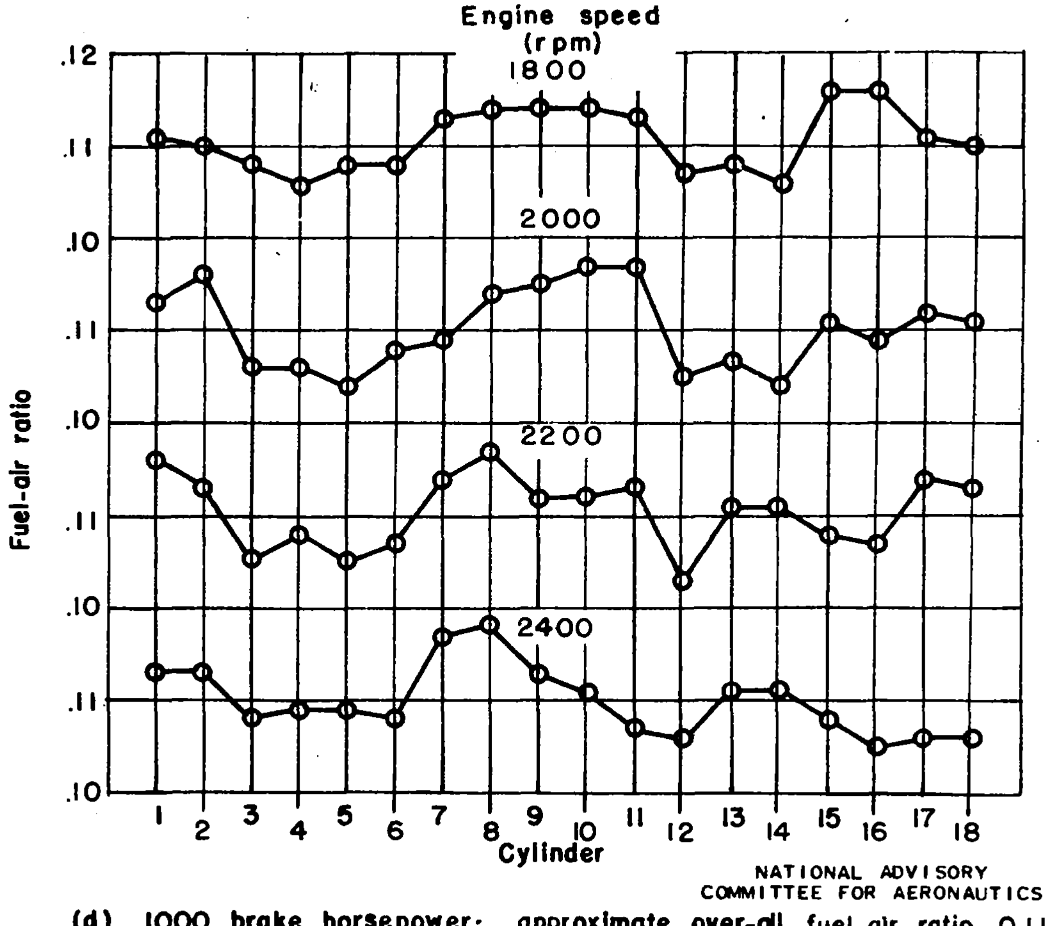
(b) 800 brake horsepower; approximate over-all fuel-air ratio, 0.106.

Figure 9.— Continued.



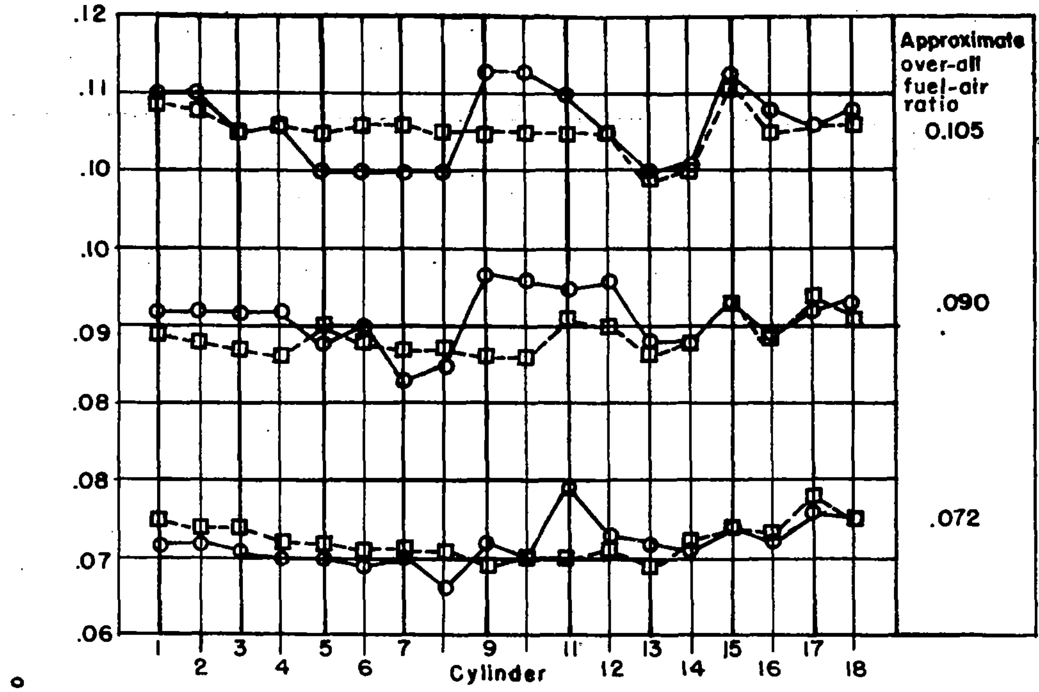
(c) 1000 brake horsepower; approximate over-all fuel-air ratio, 0.074.

Figure 9.– Continued.



(d) 1000 brake horsepower; approximate over-all fuel-air ratio, 0.110.

Figure 9.— Concluded.



(a) 800 brake horsepower.

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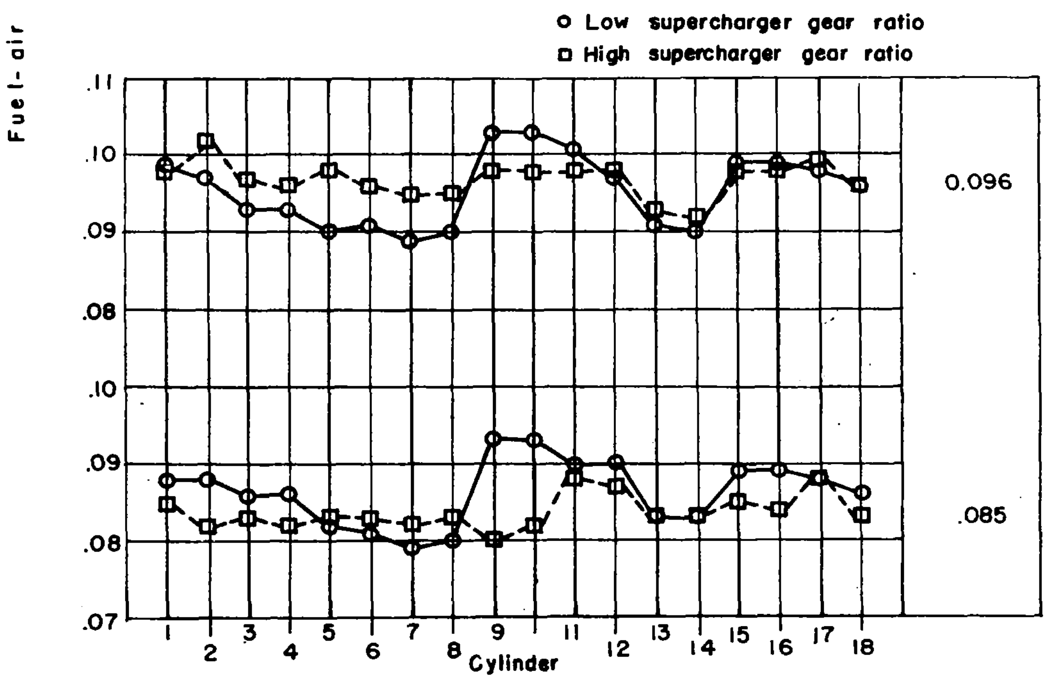
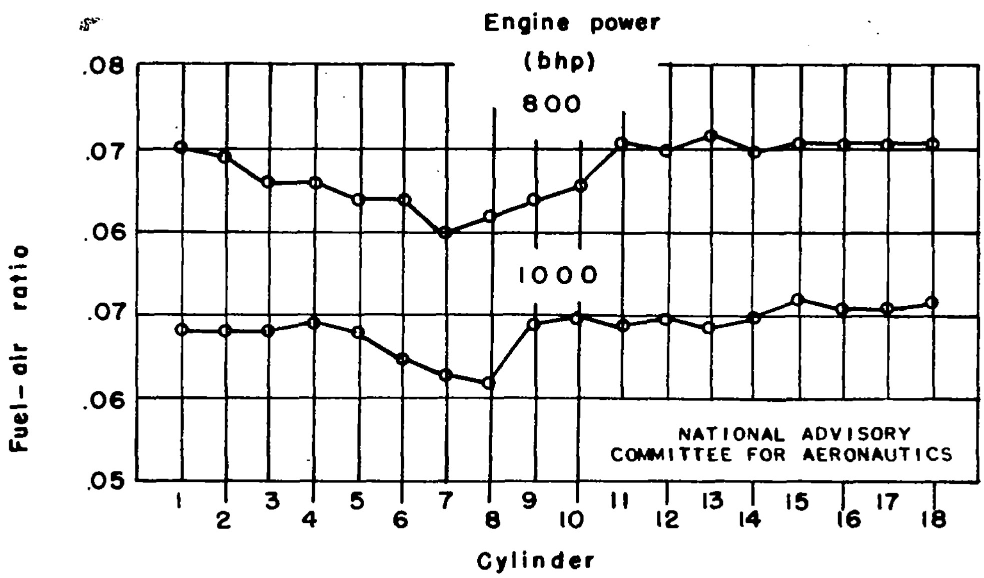


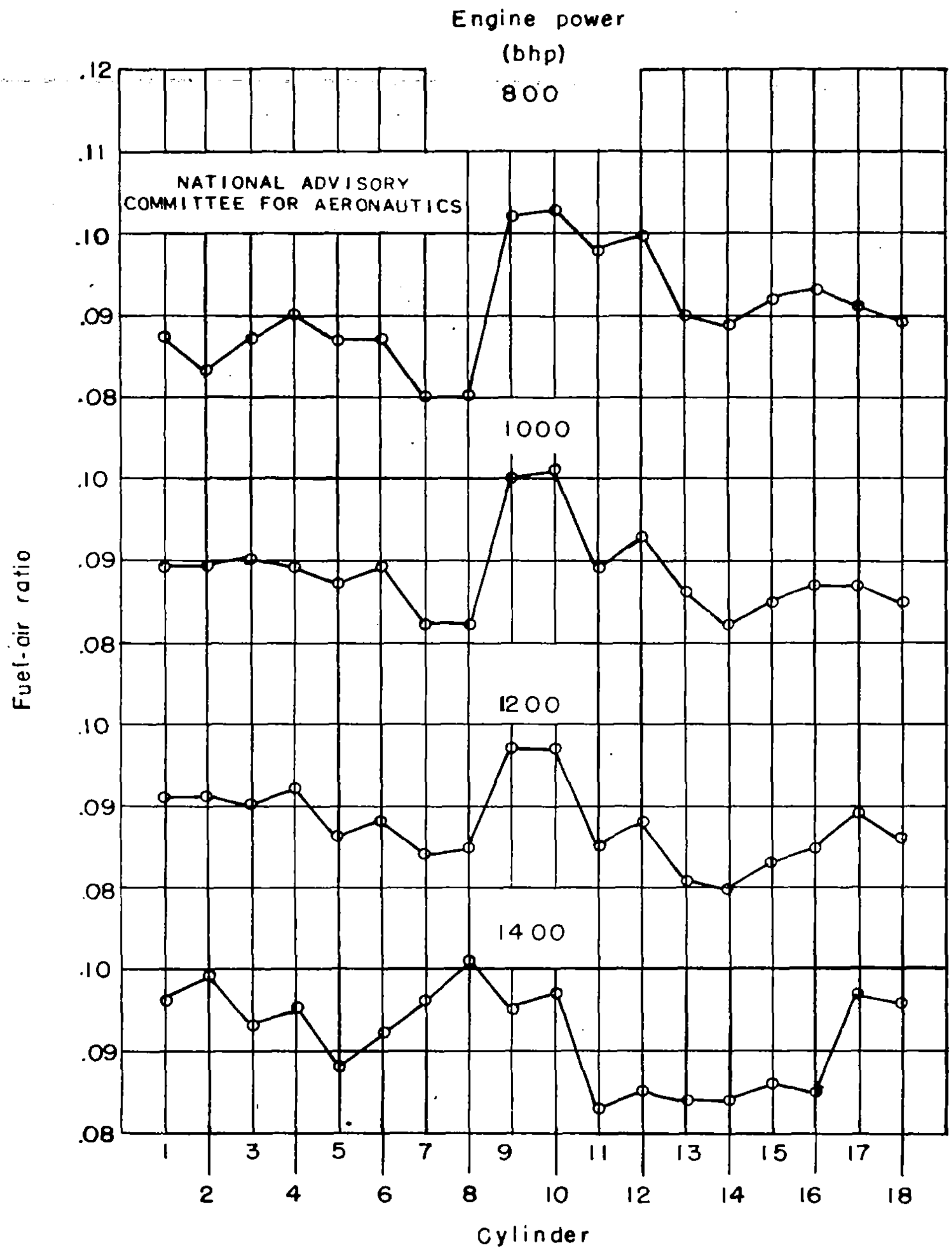
(b) 1000 brake horsepower.

Figure 10.— Effect of supercharger gear ratio on mixture distribution for various values of over-all fuel-air ratio and engine power. Engine speed, 2000 rpm.



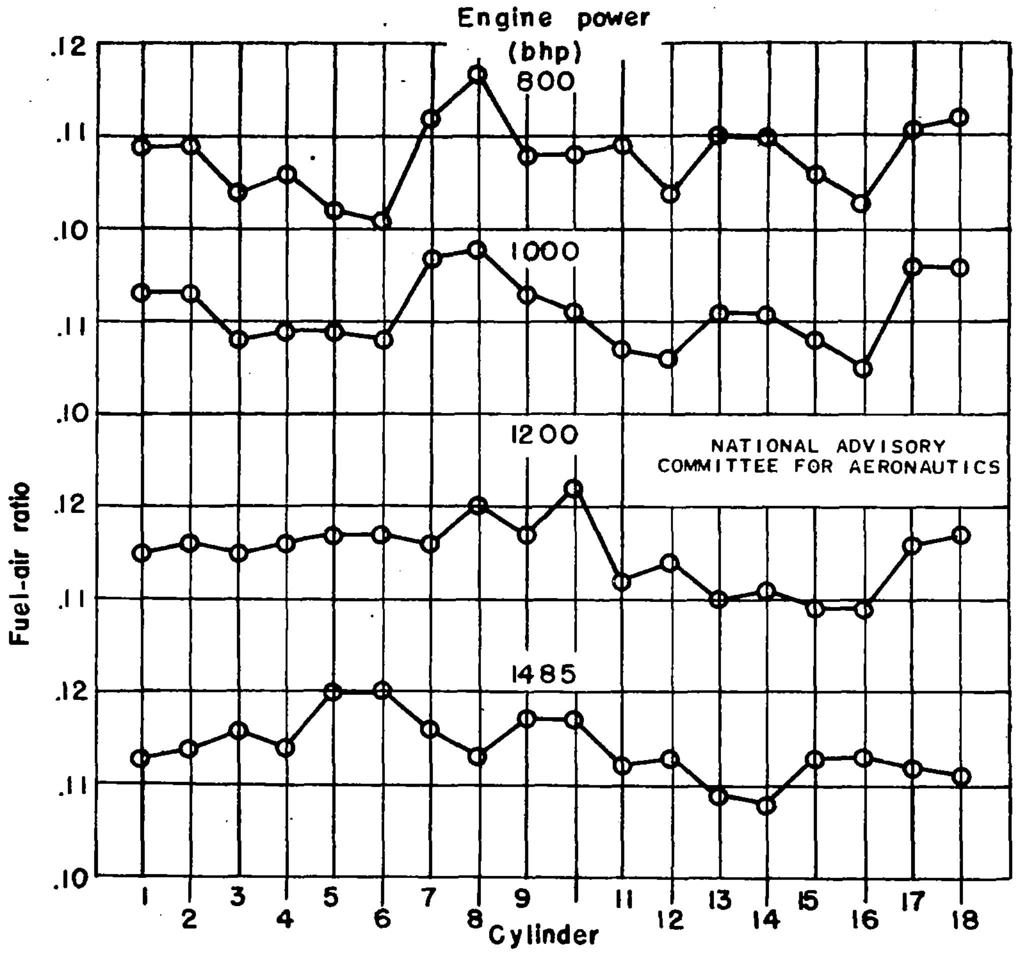
(a) Engine speed, 2200 rpm; approximate over-all fuel-air ratio, 0.068

Figure 11.– Effect of engine power on mixture distribution for various values of engine speed and over-all fuel-air ratio. Low supercharger gear ratio.



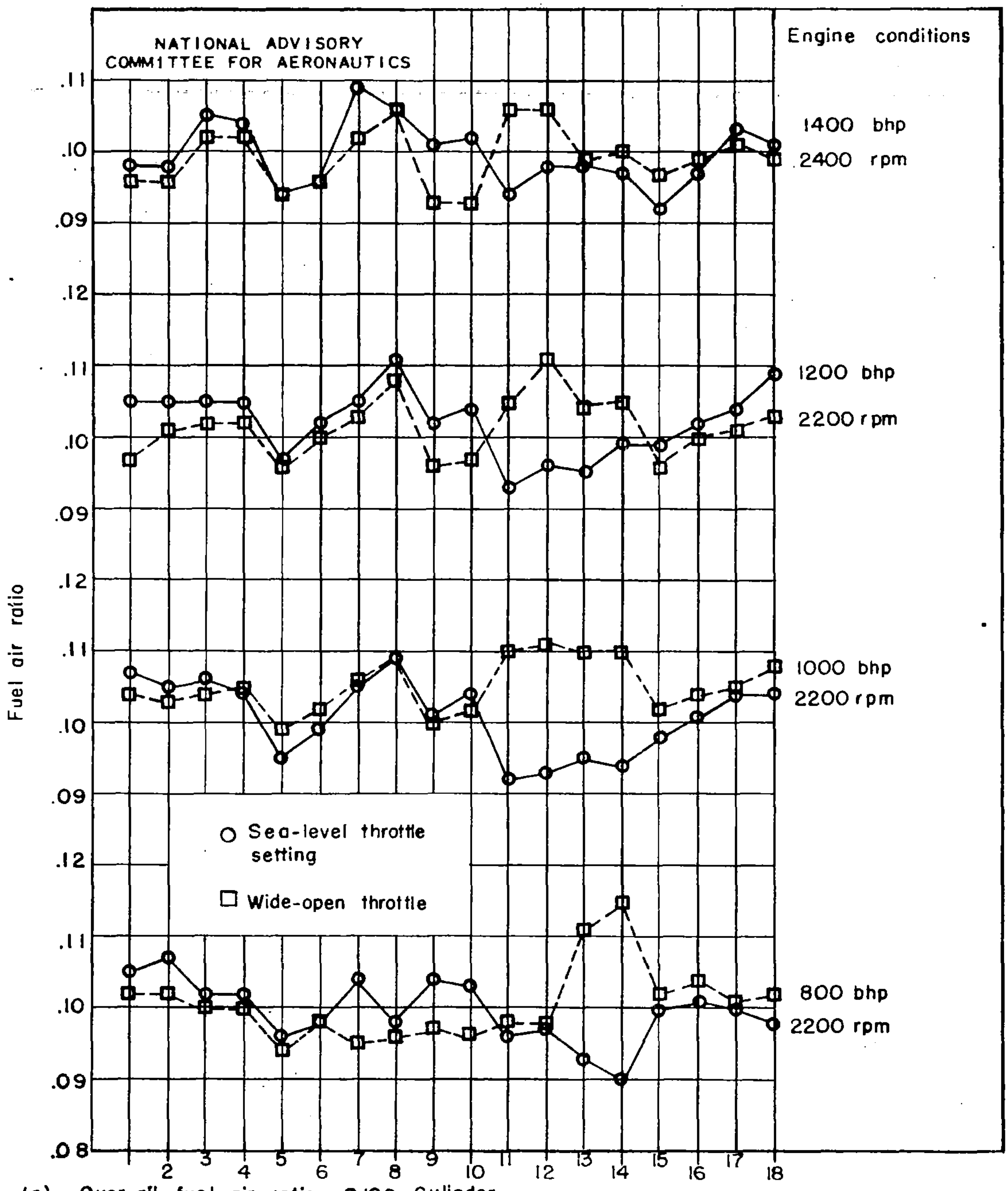
(b) Engine speed, 2200 rpm; approximate over-all fuel-air ratio, 0.089.

Figure 11.— Continued.



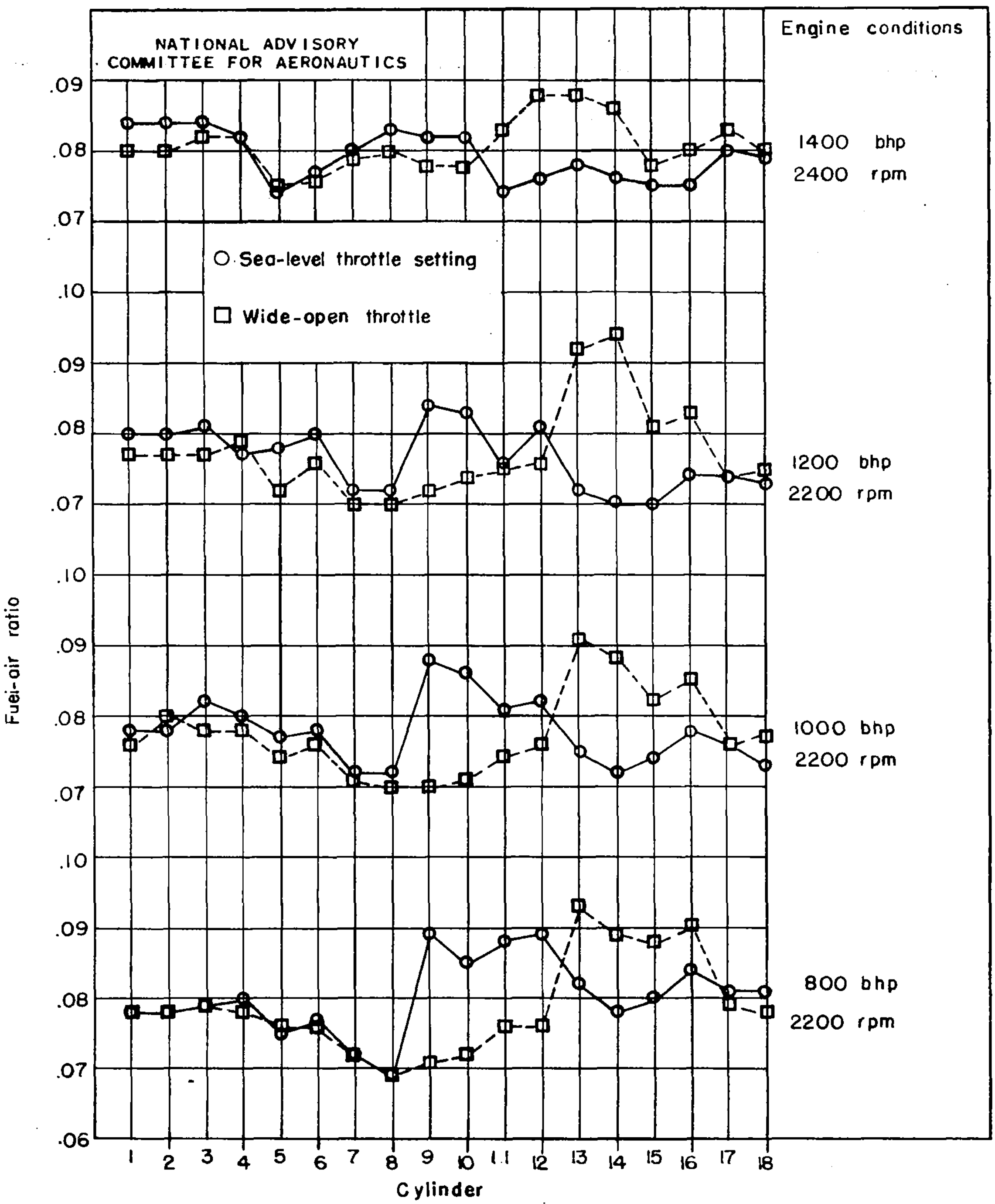
(c) Engine speed, 2400 rpm; approximate over-all fuel-air ratio, 0.113.

Figure 11.— Concluded.



(a) Over-all fuel-air ratio, 0.100.

Figure 12.— Effect of throttle setting on mixture distribution for various engine conditions and values of over-all fuel-air ratio. Low supercharger gear ratio.



(b) Over-all fuel-air ratio, 0.078.

Figure 12.— Concluded.

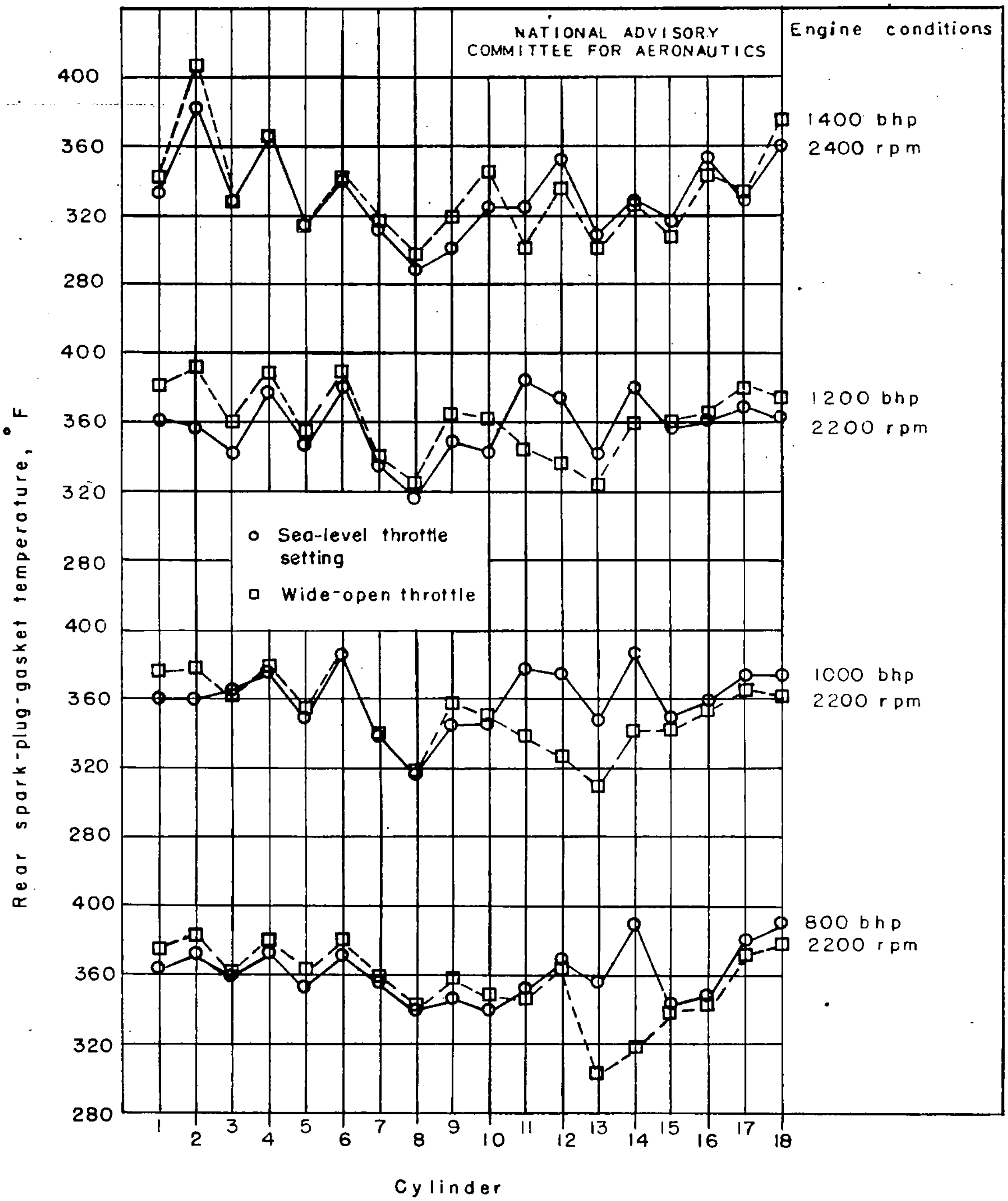


Figure 13— Effect of throttle setting on cylinder temperature distribution for various engine conditions. Low supercharger gear ratio.

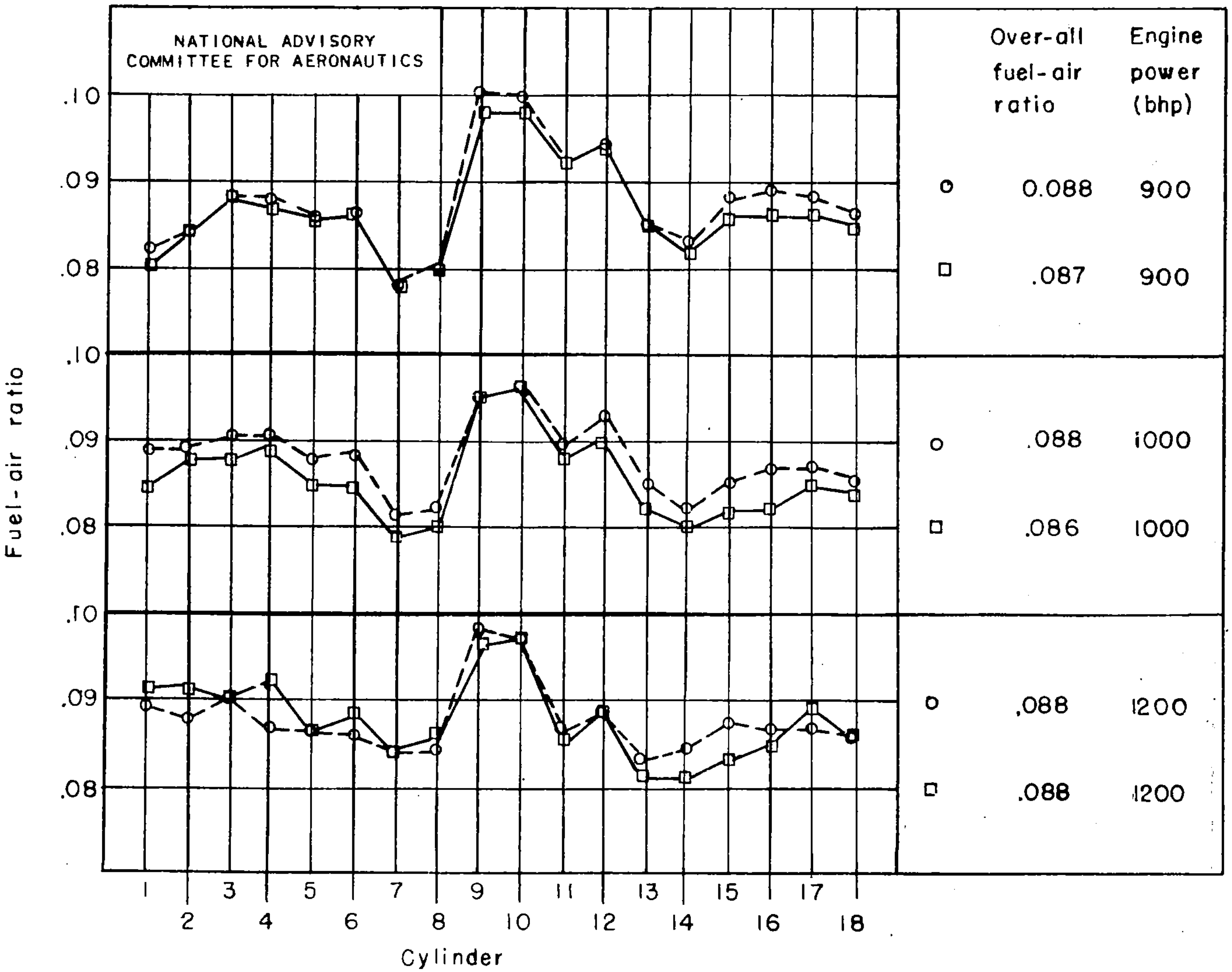


Figure 15.— Reproducibility of mixture-distribution patterns. Engine speed, 2200 rpm.

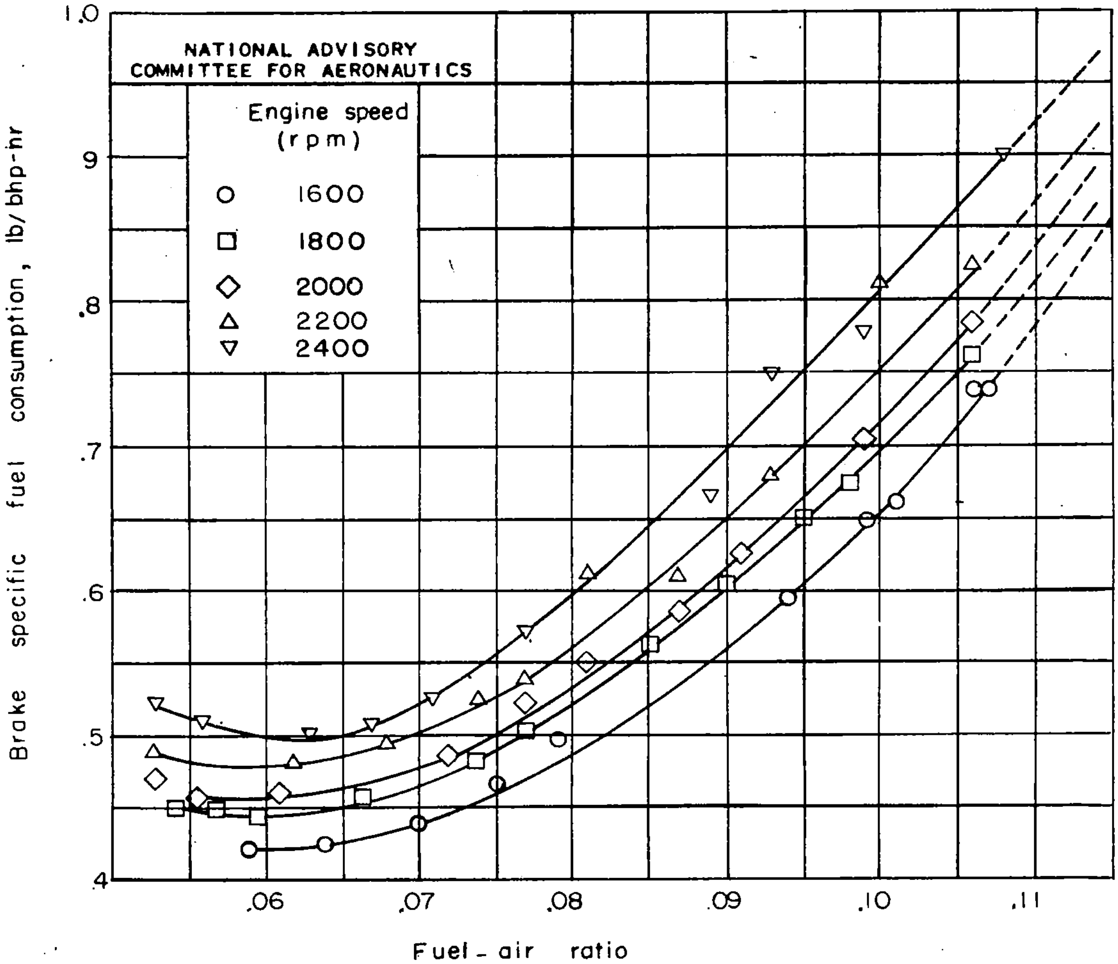


Figure 16 — Effect of fuel-air ratio on brake specific fuel consumption at various engine speeds. Engine power, 800 brake horsepower; low supercharger gear ratio.

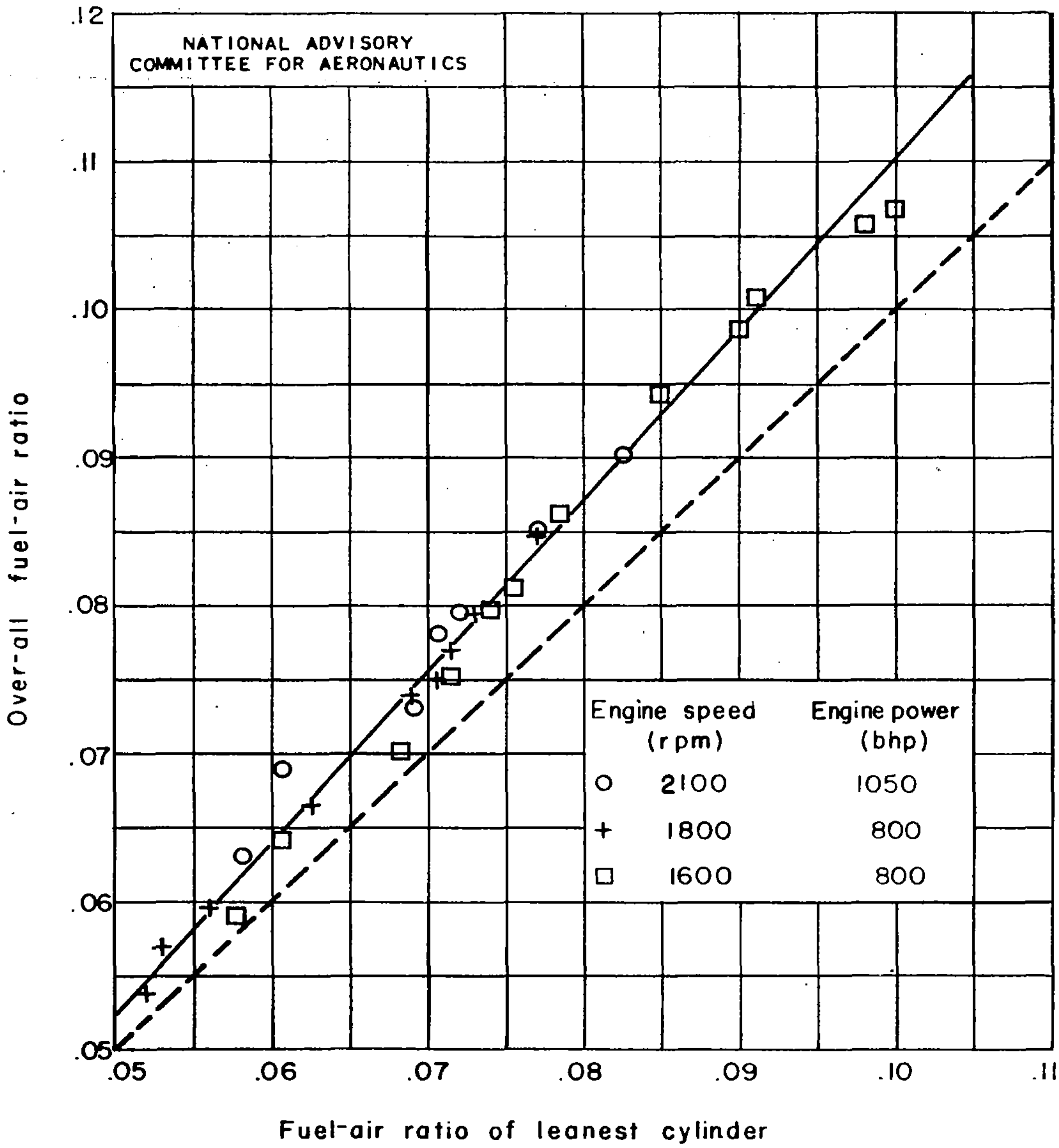


Figure 17 — Relation between the over-all fuel-air ratio and that of the leanest cylinder. Low supercharger gear ratio.

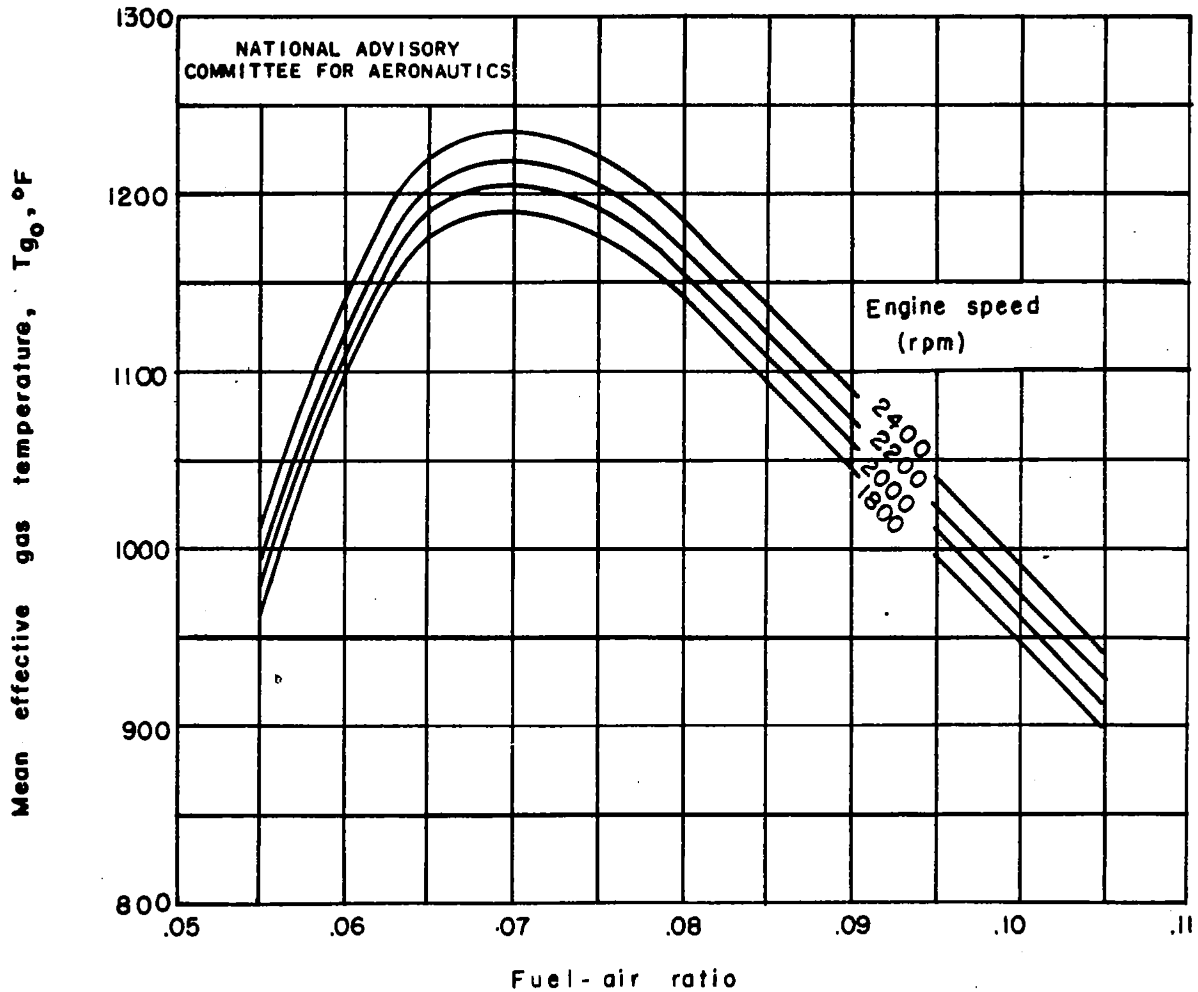


Figure 18 — Variation of mean effective gas temperature with fuel-air ratio for various engine speeds. Data from reference 9; carburetor-deck temperature, 0°F.

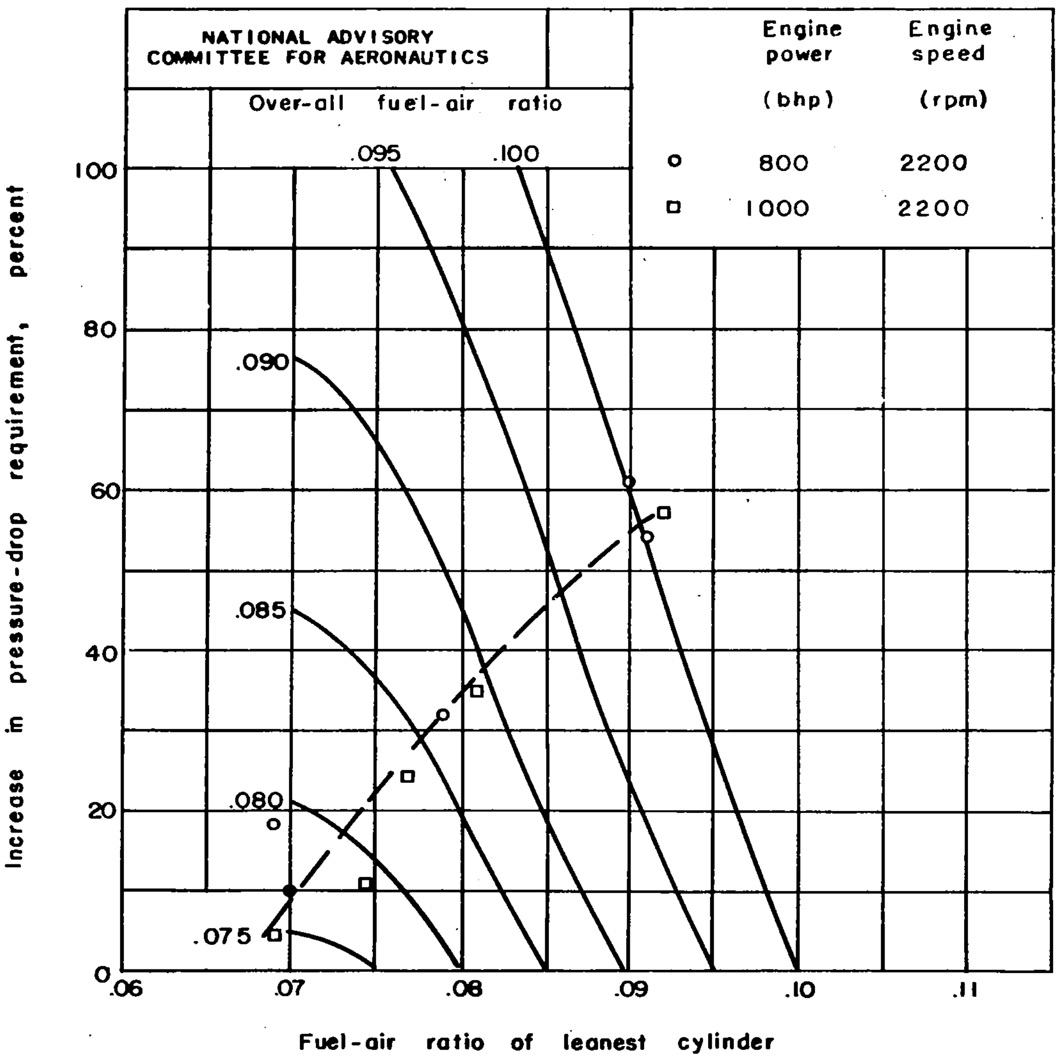


Figure 19.— Percentage increase in cooling-air pressure-drop requirement resulting from nonuniform mixture distribution. Engine speed, 2200 rpm; rear spark-plug-gasket temperature, 425 °F.